Pumps at Mines

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STEAM PUMPS

POWER AND CENTRIFUGAL PUMPS

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STEAM PUMPS

(PART 1)

Serial 3029A | Edition 1

DIRECT-ACTING STEAM PUMPS

INTRODUCTION

CLASSIFICATION OF PUMPS

1. Definitions.—A **pump,** or a pumping engine, as it is sometimes called, is a mechanical device for raising or circulating fluids, which may be liquids, such as water, or gases, such as air. A pump acts by producing a difference in pressure on the fluid in such a way that the flow takes place in a direction from the higher to the lower pressure.

Pumps are divided, according to the way they produce this inequality in pressure and cause the fluid to flow, into reciprocating, centrifugal, and displacement pumps. In a *reciprocating pump,* the fluid is moved by a piston or plunger that has a reciprocating, or to-and-fro, motion. In a *centrifugal pump,* the flow is produced by centrifugal force as the fluid is rotated very rapidly by a series of curved vanes on a revolving shaft. In a *displacement pump,* the fluid is moved by the direct action of, and contact with, steam or compressed air.

Pumps are also classified, in accordance with the power used to operate them, as steam pumps, compressed-air pumps, electric pumps, pumps driven by gasoline or oil engines, and hand pumps. Pumps may likewise be classified as *single,* which

means that there is but one pump mechanism; and as *duplex, triplex,* and *multiplex,* which means that two, three, or many (four or more), separate but similar, pump mechanisms are combined into one machine. Pumps are sometimes classified, according to the use to which they are put, as mine pumps, boiler-feed pumps, deep-well pumps, sinking pumps, etc.

2. Reciprocating Pumps.—Reciprocating pumps consist of two principal parts: a *water end* or *pump end* in which motion is imparted to the water by a piston or a plunger, and a *power end* through which the force of steam, electricity, etc., is applied to the piston or plunger in the water end. When the water is moved by a piston, as in Fig. 6, the pump is called a *piston pump,* and when it is moved by a plunger, as in Fig. 13, the pump is a *plunger pump.* If the piston or plunger moves horizontally, as in Fig. 6, the pump is a *horizontal pump,* while if the piston or plunger moves vertically, as in Fig. 1, the pump is a *vertical pump.* Reciprocating pumps are *single acting* if they deliver water at every other stroke, as in Fig. 4, and are *double acting* if they deliver water at every stroke as in Fig. 5. Reciprocating pumps may also be *tandem,* that is, have two pump cylinders arranged in line and operated by the same piston rod, as in Fig. 16.

Reciprocating pumps are divided, according to the manner in which the water piston or plunger is caused to move, into direct-acting pumps and power pumps. In a *direct-acting pump* the pump piston or plunger and the power piston are connected to the same piston rod and move back and forth together in their respective cylinders. In a *power pump* the power is communicated to the pump piston by means of belts, chains, gear-wheels, etc.

Direct-acting pumps are so very generally operated by steam that when the term *steam pump* is used, a direct-acting pump is understood to be meant. In direct-acting pumps the power end is called the *steam end* or *steam cylinder,* and the water end is often referred to as the *pump cylinder.* Steam pumps are *simple* when the steam is not used expansively, and are *compound* or *triple-expansion,* in the same sense that steam

engines are compound or triple expansion, when the steam is used expansively. Steam pumps are *condensing* or *non-condensing,* depending on whether the exhaust steam is or is not condensed; and may be single or duplex as explained in the preceding article.

Notwithstanding the fact that both power pumps and centrifugal pumps are generally operated by electric motors, the term *electric pump* is understood to mean a reciprocating power-pump actuated by electricity. Centrifugal pumps are called centrifugal pumps, or rotary pumps.

PUMP PRESSURES

3. According to the pressure they must overcome when in use, pumps may be classified as vacuum pumps, low-pressure pumps, medium-pressure pumps, high-pressure pumps, and hydraulic-pressure pumps.

4. Vacuum pumps are adapted for handling gaseous material, their field of service being in connection with condensing apparatus in power plants, tanks, heating systems, and other equipment in which it is desired to create and maintain a vacuum. The air end of vacuum pumps corresponds in its functions to the water end of steam pumps, and may sometimes require the circulation of water around it to keep down the temperature so that lubrication is possible. Cylinders arranged in this way are said to be *water jacketed.*

Vacuum pumps are extensively used in ore-dressing, metallurgical, and chemical plants in the operation of vacuum filters.

5. Low-pressure pumps are adapted for handling liquids or gases in large or small quantities against pressures not exceeding 25 pounds per square inch. Their field of service covers a great variety of commercial and industrial enterprises, water pumps of this class and of small capacity being found in heating and refrigerating plants, tank service, in factories, in mines, and in contractors' use. Units of larger capacity up to 60,000 gallons a minute are used in the condenser systems

of large power plants, in sewage, drainage, and irrigation projects, in the operation of dry docks, locks, etc. The air and gas pumps used at blast furnaces and gas works are of the low-pressure type.

6. **Medium-pressure pumps** are adapted to work against pressures not exceeding 125 pounds per square inch, and have a broader field of service than any of the preceding or following classes. Medium-pressure water pumps of small capacity are used for feeding boilers, for draining mines, for water supply, etc., and similar units of larger capacity up to about 10,000 gallons a minute are found in water-works stations and the like. Air compressors, displacement pumps, and the air pumps used in connection with the mechanical brakes on steam and electric railroads, are of the medium-pressure type.

7. **High-pressure pumps** have a more limited field of service than the preceding classes. They are adapted to handling small or medium quantities of liquids or gases against pressures up to 500 pounds per square inch. Pumps of this class and of small capacity are adapted for boiler feeding, forcing oil into bearings subject to great pressure, for elevators, accumulator tanks, etc. Units of medium capacity up to 1,500 gallons a minute are found in use at deep mines, in high-pressure fire service, cross-country oil lines, etc. High-pressure air pumps are not usually of very large capacity. They are used in connection with ammonia and refrigerating plants, in the operation of hydraulic gates, and in storage and accumulator tanks at chemical works.

8. **Hydraulic-pressure pumps** are a specialty. They are usually constructed to handle from 100 to 500 gallons of water a minute against pressures that may sometimes reach 3,000 pounds per square inch. They are used in handling liquids in connection with metal-shearing, shaping, and riveting machines, hydraulic presses, etc. Air and gas pumps of this class are found in service at chemical, liquid air, and refrigerating plants, and at power stations, supplying air to compressed-air locomotives.

POWER FOR PUMPING

9. Steam Power.—Where steam is available, it may be employed with advantage in operating pumps. The cost of a steam plant is lower than that for any other kind of power, except in special cases, but there is a loss of power, particularly in cold weather, due to the condensation of the steam when transmitted long distances in pipes from the boiler to the pump. The condensation of steam may be prevented, to a great extent, by the use of pipe coverings made of asbestos, magnesia, or other materials that are poor conductors of heat. The use of pipe coverings, however, adds to the first cost of the plant and makes it difficult to locate any leak that may occur without removing a considerable portion of the covering. When power has to be transmitted any great distance, it will generally prove economical to use electrically-driven power or centrifugal pumps instead of steam pumps; and this is particularly true at plants where electricity is used for operating other machinery.

In mining practice, the disposal of the exhaust steam is not always a simple problem. Several methods of disposing of it are in use; it is sometimes conveyed to the surface through a pipe laid in the entry or a drill hole sunk for that purpose; it is discharged directly into the upcast shaft; it may be led into the sump where it is condensed, or a condenser may be used. When the exhaust is conveyed to the surface through a pipe of considerable length, trouble is caused by the condensation of steam in the pipe, and by the heat radiated in the mine passages. The friction due to a long exhaust pipe decreases the efficiency of the pump by increasing the back pressure. The practice of discharging the exhaust steam into the upcast is often ruinous to the shaft timbering. When the pump exhausts into the sump, it is often found that the whole body of water is heated to a comparatively high temperature, and the temperature of the mine and humidity of the air are increased to such an extent that the mine timber decays rapidly. At times, also, the roof on the airways and traveling ways is softened and becomes troublesome and dangerous. A partial remedy for this is to condense the steam in a con-

denser. When this is done and the water of condensation is drained into the sump to be pumped again to the surface, much of the trouble due to exhaust steam is overcome, but the heat from the steam pipes supplying the pump remains.

It should also be noted that steam pumps are practically useless if they become submerged.

10. Compressed-Air Power.—Compressed air as a power for driving pumps is usually limited to plants where it is used for operating other machines. Thus, where compressed air is used for driving rock drills as in quarrying, tunneling, shaft sinking, mining, excavating for foundations, and similar undertakings, it may be used economically to drive the pumps. Under the conditions mentioned, compressed air has several advantages over steam. There is small loss of pressure in transmission even where the piping is long; air coming from the pumps or leaking from the piping assists the ventilation; and the low temperature caused by the expansion of the air keeps the station cool. And, above all, a compressed-air pump will work when entirely submerged under water, while a steam pump will not. Under ordinary conditions, however, it is not economical to install an air compressor for driving pumps, because of the increased cost of the plant, and because of the low efficiency, unless the air is heated before being used in the power cylinder of the pump.

Compared with electricity, compressed air or steam are less dangerous in coal mines, because neither of them can ignite gas or dust, whereas both gas and dust can be ignited by the sparks from an unprotected electric motor. However, the use of alternating current motors or of explosion-proof direct-current motors has practically eliminated the dangers at one time connected with the use of electricity.

11. Electric Power.—The electrical system of power transmission for pumping is, in many ways, an ideal one, owing to the ease and cheapness of its installation; the practical absence of any limitations of the distance to which it may be transmitted; freedom from the disagreeable features connected with the use of steam; low transmission losses; and general

high efficiency. The electric current may be purchased from central-station service lines or can be generated at the most convenient point, using either water-power or a steam engine to drive the dynamo.

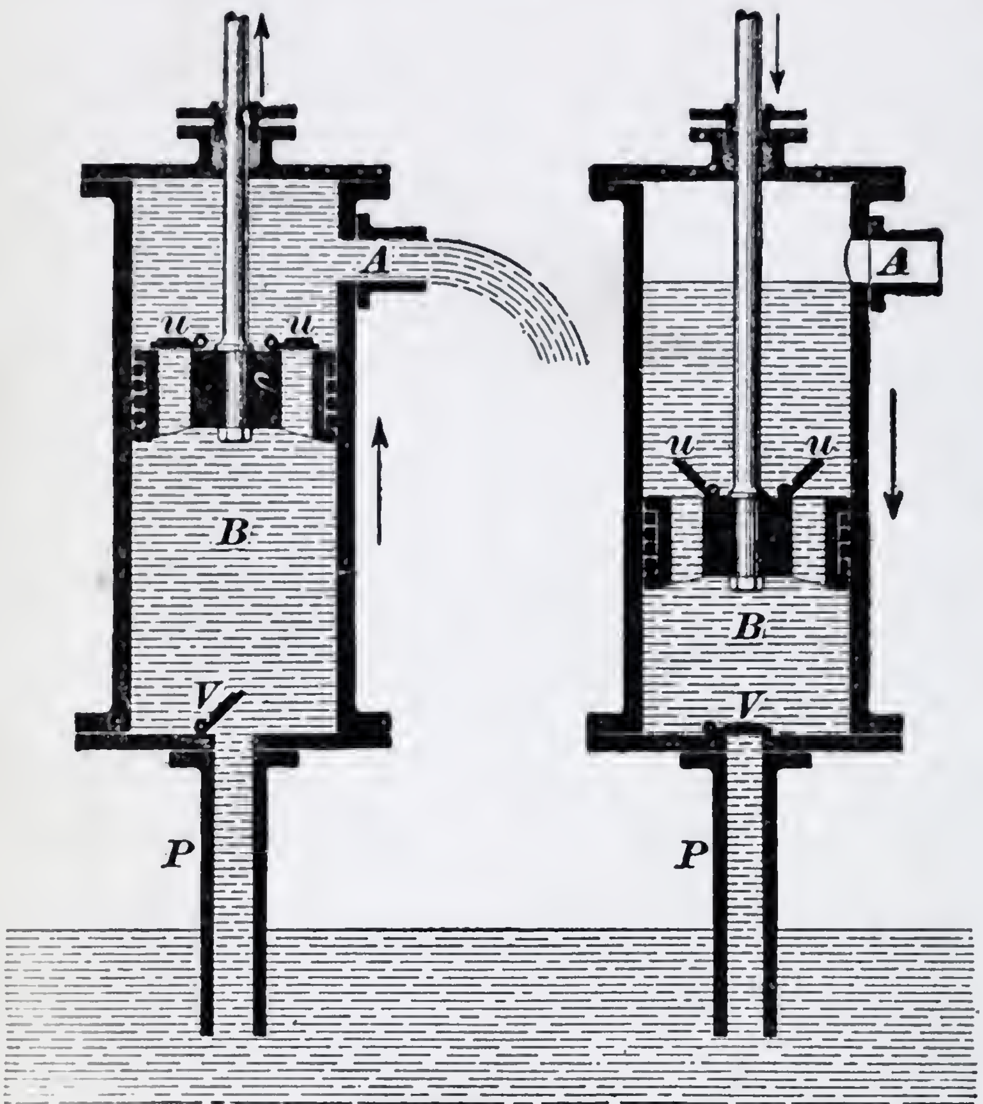
Often, auxiliary pumps are required in places where the use of steam would be prohibitive and compressed air would be expensive. The ease with which an electrically driven pump mounted on a truck can be moved to such places and the wires for the transmission of power placed in position appeals to every one. Electricity is now so generally used as a power in mines, mills, factories, and engineering operations of all kinds, that electric pumps, both power and centrifugal, are very extensively used and are cheaply and efficiently handling large volumes of water against heavy pressures.

12. Gas and Oil Engines.—Where natural gas, gasoline, or heavy oils are abundant and cheaper than coal, they may be used in properly designed engines for driving pumps. The engine may be direct-connected to the pump, or power may be transmitted from the engine to the pump by belts, chain, or gears; or the engine may be used to drive an electric generator to supply current for the pumps. An installation of the latter kind is very reliable and, if properly taken care of, renders satisfactory service from every point of view.

ACTION OF PUMPS

13. Suction Pumps.—In Fig. 1 (*a*) is shown a cylinder *B* in which travels a closely fitting piston; and connected to the cylinder is a pipe *P*, one end of which is immersed in the water of a well. If, then, the piston is drawn upwards, as shown in the illustration, the pressure of the atmosphere on the surface of the water in the well will cause the water to rise in the pipe, lift the valve *V*, and fill the space in the cylinder emptied by the piston. The height to which a column of water may be raised by atmospheric pressure at sea level is, theoretically, about 34 feet, but, for reasons that will be given later, this theoretical *height of suction*, as it is called, is never reached in practice.

As the piston descends, as shown in Fig. 1 (*b*), the weight of the water in the cylinder closes the valve *V* and prevents the escape of water through the pipe *P*. The downward motion of the piston tends to compress the water in the cylinder, but this is prevented by the opening of the valves *u* in the piston, and when the latter has reached the end of its stroke practically all the water in the cylinder is above the piston. When the piston begins its up stroke, the weight of the water above the piston closes the valves *u*, preventing the water above the piston from flowing back into the lower part of *B*. As the piston rises in the pump barrel, the space *B* is again filled with water, which flows in through the pipe *P* and the valve *V*, while the water that is above the piston is lifted with it and discharged through the spout *A*. A simple pump of this form, in which there is no valve in the pipe above the piston, and which lifts the water only to a point just above the upward travel of the piston, is often called a *suction pump,* in distinction from the pump shown in Fig. 2, in which the water is lifted to a greater height.



(*a*) FIG. 1 (*b*)

14. Lift Pumps.—If a valve *c*, Fig. 2, is placed in the delivery pipe *P′* above the piston, it will open when the piston is on its upward stroke and permit the passage of water, but will close when the piston begins its downward stroke and thus prevent the water from flowing back into the upper part of the cylinder *B*. The water may be discharged through the cock

shown just above *c*; or, if this is closed, it may be lifted to any desired height, since each upward stroke of the piston raises more water into the pipe *P′*. A pump of this kind is sometimes called a *lift* or *lifting pump,* but it is evident that there is no difference in principle between the pumps shown in Figs. 1 and 2, and that they are both suction and lift pumps.

The cylinder *B,* Figs. 1, 2, and 3, is called the *working barrel* or *pump cylinder,* the pipe *P* through which the water is drawn into the pump cylinder is the *suction* or *tail pipe,* and the pipe *P′* is the *delivery, discharge,* or *column pipe.* The reservoir from which the water is drawn through the suction pipe *P* is the *well* or *sump.* At *V* is the suction valve, at *c,* Fig. 2, or *V′* in Fig. 3, is the discharge valve, and at *u* are the piston valves. At *S,* Fig. 2, is a stuffingbox to prevent the escape of the water where the piston rod passes through the head of the pump barrel.

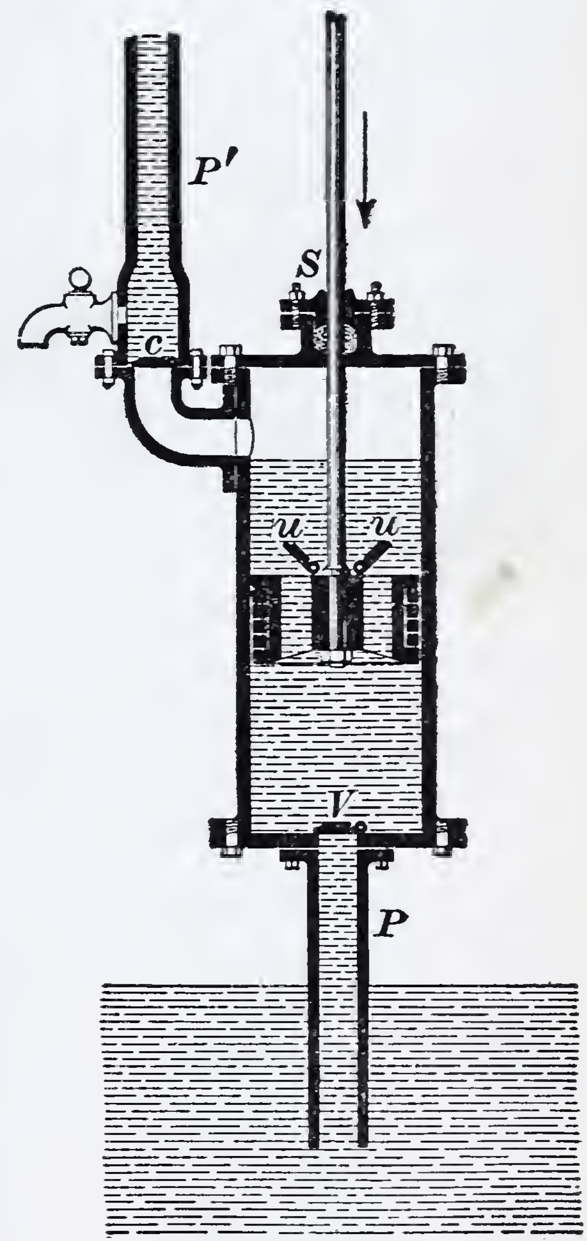


FIG. 2

15. Force Pumps.—Water can be raised by a lift pump to a considerable height; but since the entire weight of the water comes on the valve *c,* Fig. 2, a pump that works on the principle illustrated in Fig. 3 is generally used where water has to be raised to a great height. This type of pump is known as a *force pump* because the water is forced up the delivery pipe by the downward stroke of the piston instead of being lifted into it on the upward stroke of the piston as in Fig. 2. The force pump shown in Fig. 3 is also known as a piston pump because the water is moved by a solid, valveless piston that fits closely into the cylinder. The water is drawn up the suction pipe *P* when the piston rises; but when the direction of the motion of the piston is reversed, the pressure of the water caused by the descent of the piston closes the valve *V,*

opens the valve V', and forces the water up the delivery pipe P'. When the piston again begins its upward movement, the valve V' is closed by the pressure of the water above it and the valve V is opened by the pressure of the atmosphere on the water below it, just as in the lift pump. A stuffingbox is unnecessary with a single-acting piston pump.

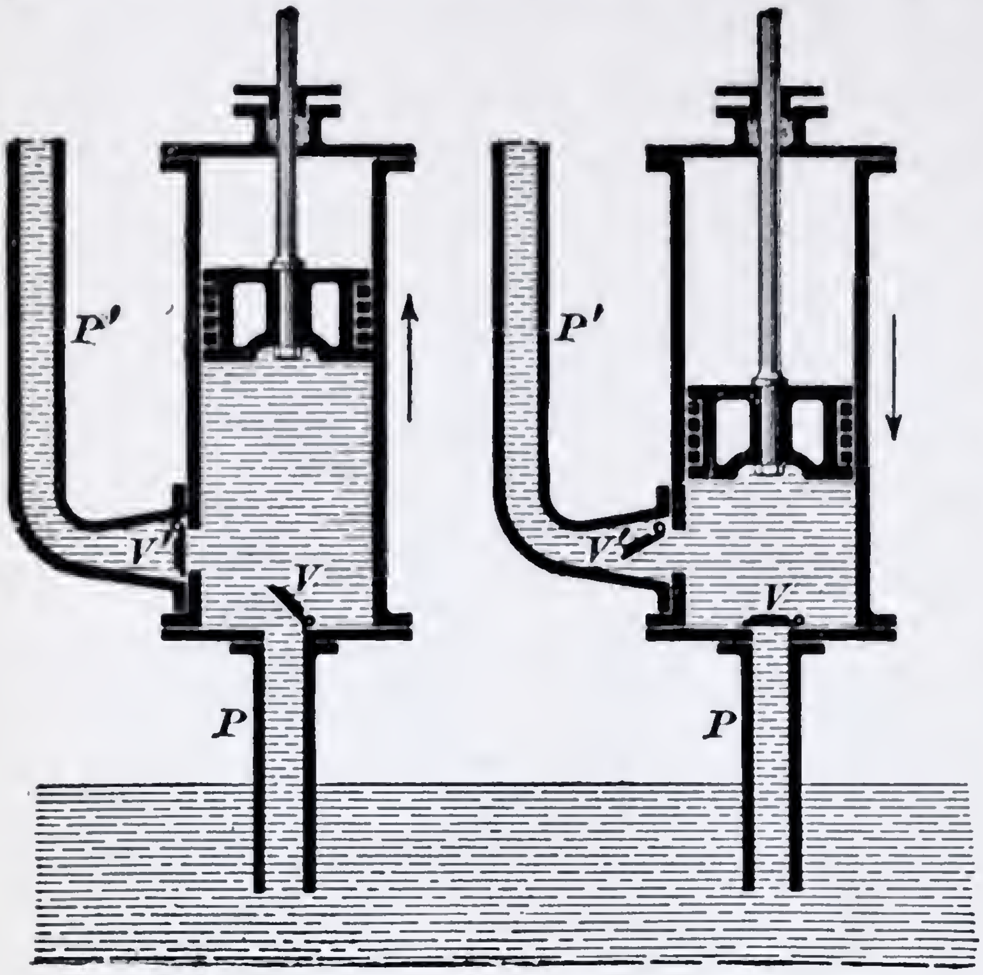


FIG. 3

16. If a force pump such as is shown in Fig. 3 is used to force water to a great height or against heavy pressure, the pressure makes it very difficult to keep the water from leaking past the piston, and the piston must be constantly packed and repaired, which necessitates shutting down the pump. To overcome this disadvantage when pumping against heavy pressures, the piston of the pump is often replaced by a *plunger* which consists of a solid or hollow closed cylinder that nearly fills the pump barrel in which it works to and fro as

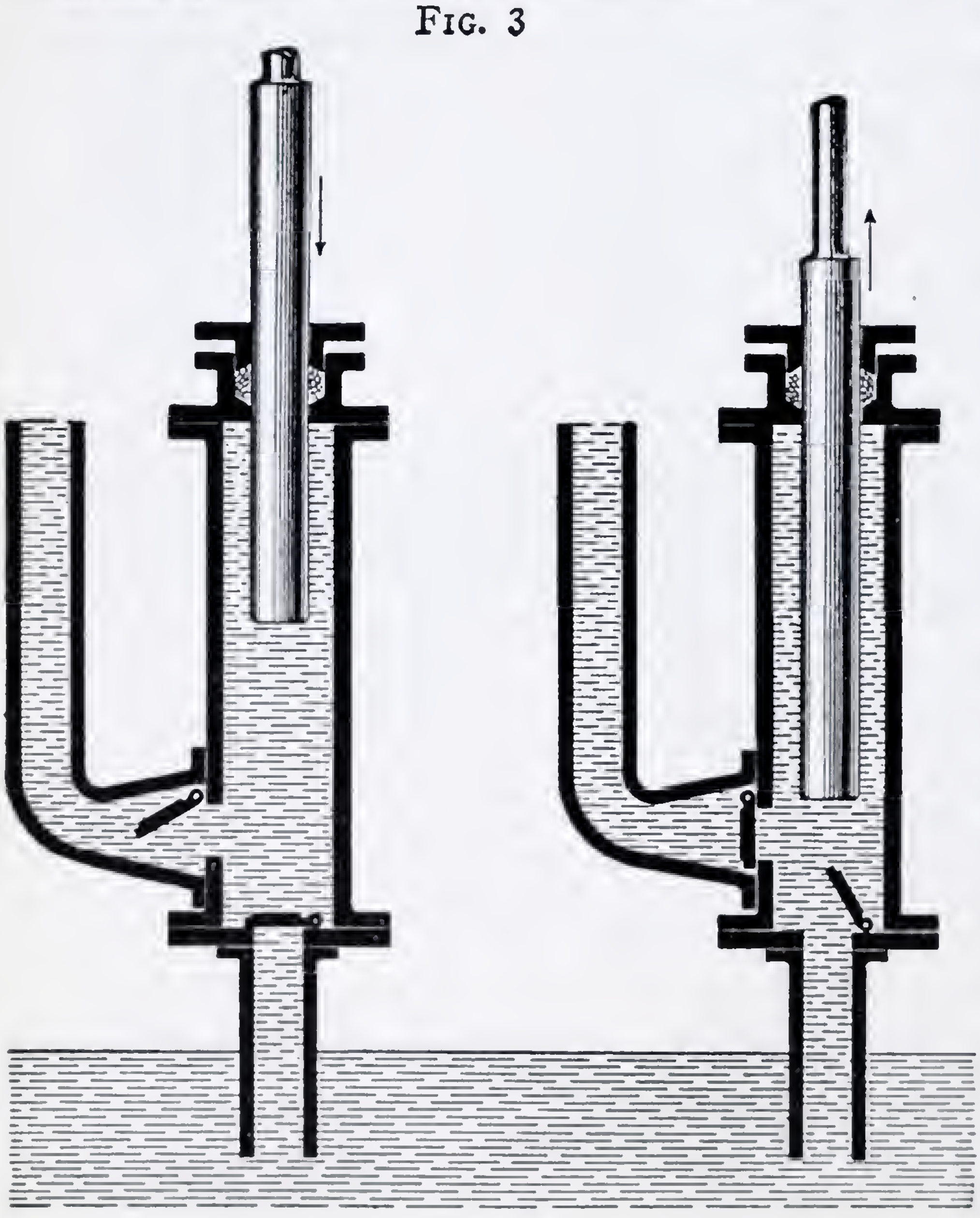


FIG. 4

shown in Fig. 4. A force pump of this type is often called a *plunger pump,* and is provided with a stuffingbox where the plunger passes through the head of the pump cylinder which is more easily packed than the piston shown in Fig. 3. The action of the plunger on its up stroke is precisely the same as the piston in the pumps already explained; but, as the plunger descends, it forces through the delivery valve an amount of water equal in volume to the displacement of the plunger.

The force pumps shown in Figs. 3 and 4 are single-acting; that is, the water is forced into the delivery pipe during the down or forward stroke of the piston or plunger. Force pumps, of either the piston or plunger pattern, may be constructed so as to force water into the delivery pipe during both the forward and the return stroke. They are then called double acting, and are used wherever a nearly continuous discharge is necessary. To discharge a given quantity of water, more power is required with a single-acting than with a double-acting pump, and the shocks on the pump and piping are also much greater.

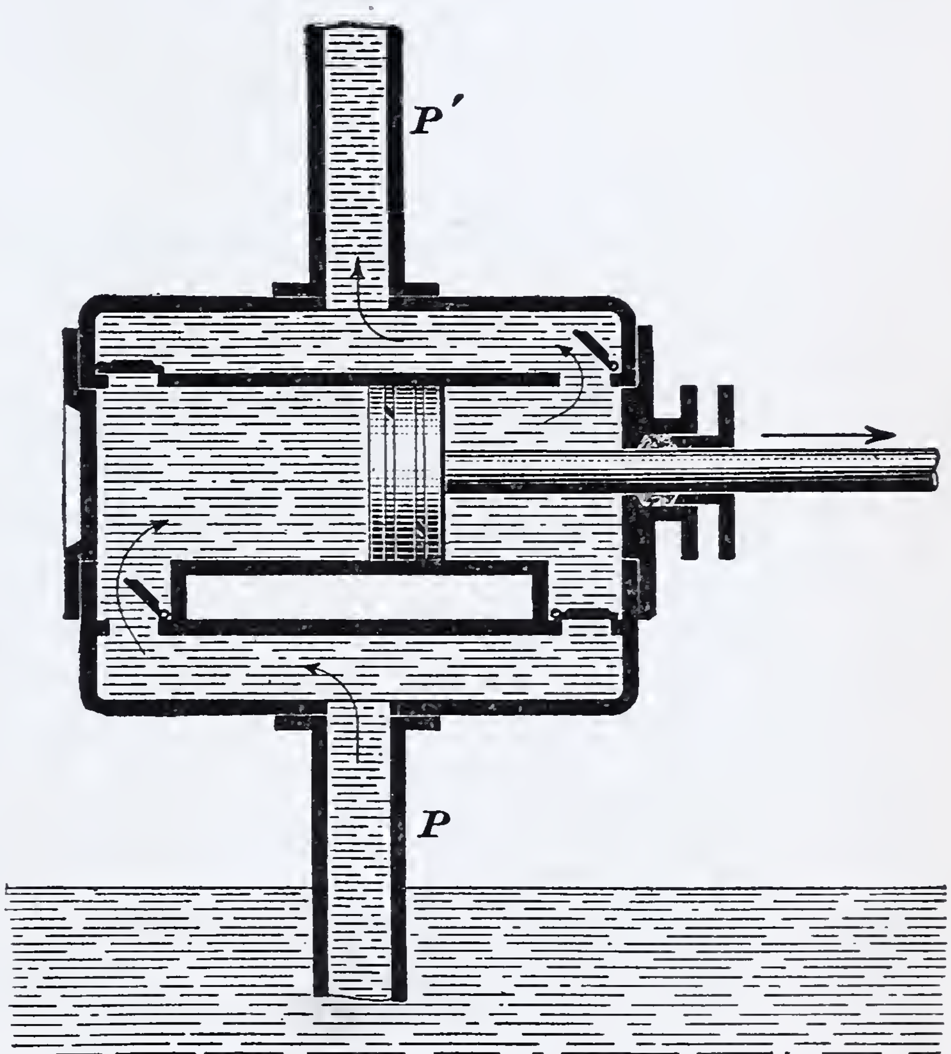


Fig. 5

17. A double-acting force pump of the piston pattern is shown diagrammatically in Fig. 5. Such a pump has two sets of suction valves, one set for each side of the piston. With the piston moving in the direction of the arrow, the pressure of the atmosphere forces the water up the suction pipe P into the left-hand end of the pump cylinder, the left-hand suction valve opens, and the left-hand delivery valve closes. The piston, in moving to the right, displaces the water in the right-

hand end of the pump cylinder; as a consequence, the right-hand suction valve is closed and the right-hand delivery valve opened. The water now flows up the delivery pipe P'. Imagine that the piston is at the end of its stroke and commences to move to the left. Its first movement promptly closes the left-hand suction valve and opens the left-hand delivery valve. It also closes the right-hand delivery valve and opens the right-hand suction valve. It is thus seen that with the arrangement given the piston will discharge water both during the forward and return stroke. While the pump shown is a horizontal pump, the same principle may be applied to a vertical pump.

TYPES OF STEAM PUMPS

SIMPLE PUMPS

18. Single Direct-Acting Pumps.—The single direct-acting pump is one of the most common forms of steam pump. It is a pump in which there are no revolving parts, the pressure of the steam in the steam cylinder being transferred to the piston or plunger of the pump in a direct line by means of a continuous rod or connection. In pumps of this construction, the moving parts have no weight greater than that required to produce sufficient strength in such parts for the work they are expected to perform; as there is, consequently, no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off steam in the cylinder during any part of the stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a uniform pressure of steam throughout the entire stroke of the steam piston against a uniform resistance of water pressure in the pump; the difference between the power exerted in the steam cylinders over the resistance in the pump governs the rate of speed at which the piston or plunger of the pump will move.

A single direct-acting pump is usually low in first cost and simple in construction, but the action of the pump plunger

or piston is an intermittent one, producing an irregular flow of water and subjecting the pump and connecting pipes and their joints to severe strains, especially when pumping under heavy pressures.

19. Double-Acting Piston Pump.—A *horizontal, single, simple, non-condensing, double-acting, piston pump,* is shown in section in Fig. 6. It is a horizontal pump because the recip-

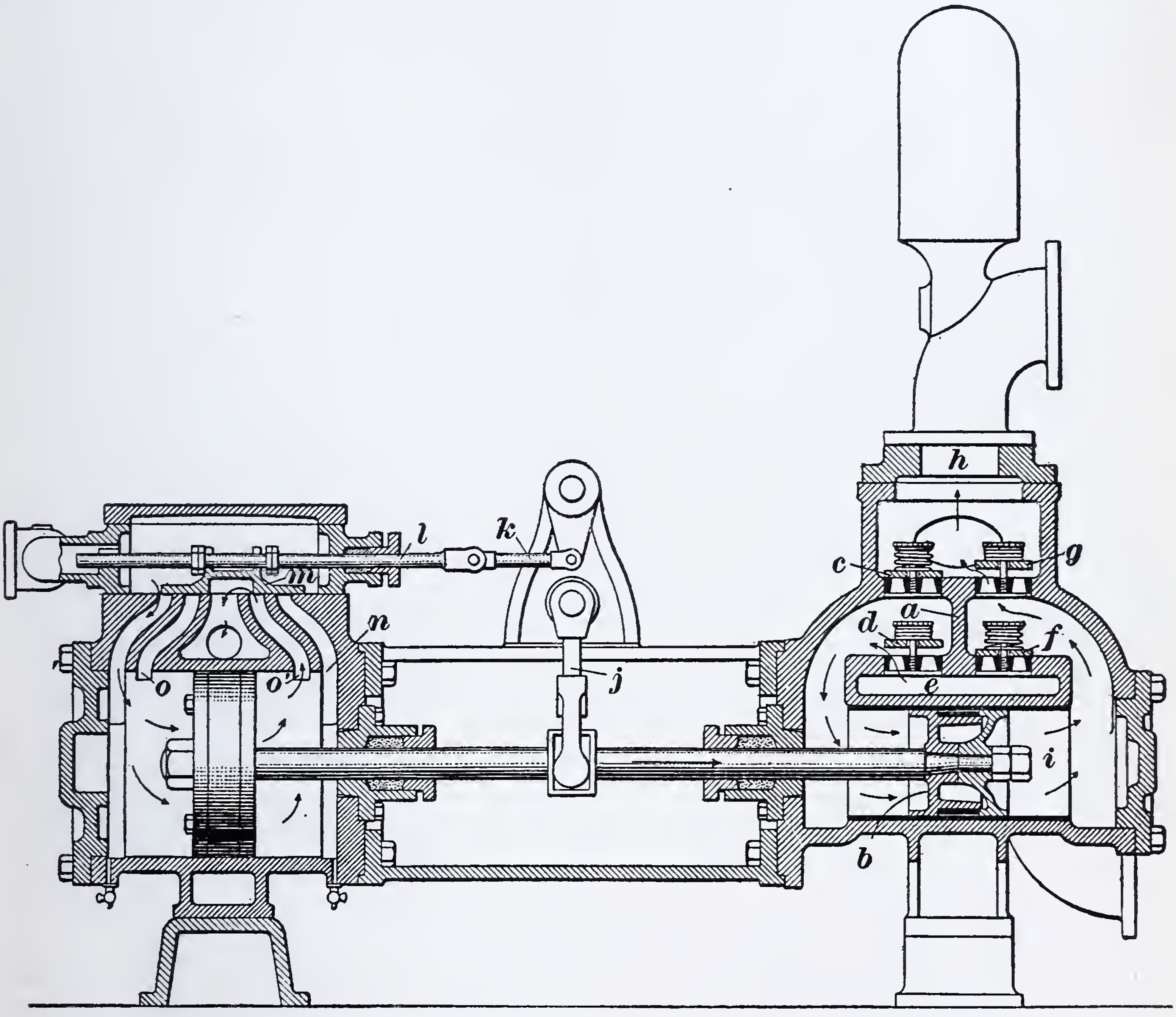


Fig. 6

rocating parts move horizontally; a single pump because there is but one pump mechanism consisting of a steam and a water cylinder; simple, because the steam is not used expansively; non-condensing, because the exhaust steam is not condensed; double acting, because it both draws in and discharges water on each stroke; and piston, because the water is moved by a piston.

The water end is divided into practically two compartments above the cylinder by the partition *a,* each compartment being

supplied with an inlet and a discharge valve. When the piston *b* moves to the right, the discharge valve *c* closes, while the inlet valve *d* opens and permits water to enter from the suction pipe *e* to fill the space back of the piston; at the same time, the inlet valve *f* is closed and the discharge valve *g* is opened, and the water is forced through it by the piston and out at *h*. On the return stroke, the discharge valve *g* is closed and the inlet valve *f* is opened and water drawn into the front end of the cylinder *i*. The pressure of the water in the back end of *i* now closes the inlet valve *d*, opens the discharge valve *c*, and the water discharges through the opening *h*.

The piston rod, which is connected to both steam and water pistons, is arranged to throw the steam-valve mechanism as it moves by means of levers *j*, *k* that work the valve rod *l* attached to the **D** valve *m*. In the illustration, the pump is just beginning its forward stroke, the steam port *n* and the exhaust port *o* being closed, while the steam port at the other end of the cylinder and the exhaust port *o'* are just opening. It will be noticed that the steam piston is considerably larger in diameter than the water piston in order to overcome the water pressure.

20. Another form of piston pump, differing somewhat in the water end and considerably in the steam end from that shown in Fig. 6, is illustrated in Fig. 7. The pump consists of a water cylinder *a* and a steam cylinder *b* placed on the same axis and united by a cast-iron yoke *c*. This yoke is provided with projections *d* that enter counterbores on the ends of both cylinders and form cylinder heads. Except in the cheapest grades of pumps, the steam cylinders, the water cylinders, and the yoke are cast separately, because the water cylinder is liable to burst from freezing or other causes. If the cylinder is cast in one piece, bursting will ruin the entire pump; but if the parts are cast separately, the broken part can be replaced. The water piston *e* and the steam piston *f* are attached to the same piston rod *g*. When the piston has nearly completed its stroke, the steam in the steam chest *h* sets in motion an auxiliary piston valve *i*, which in turn shifts the

main slide valve j and opens and closes the steam ports k leading to the cylinder b and the exhaust port l leading from the cylinder to the atmosphere. Because the auxiliary valve is moved by steam pressure instead of by positively operated connecting mechanism, as in Fig. 6, it is called a *steam-thrown valve*. The valve chamber m of the water end is fitted with suction valves n and delivery valves o. An air chamber p is attached to the valve chamber as near the pump as possible, so as to provide a cushion for the water as it passes from the

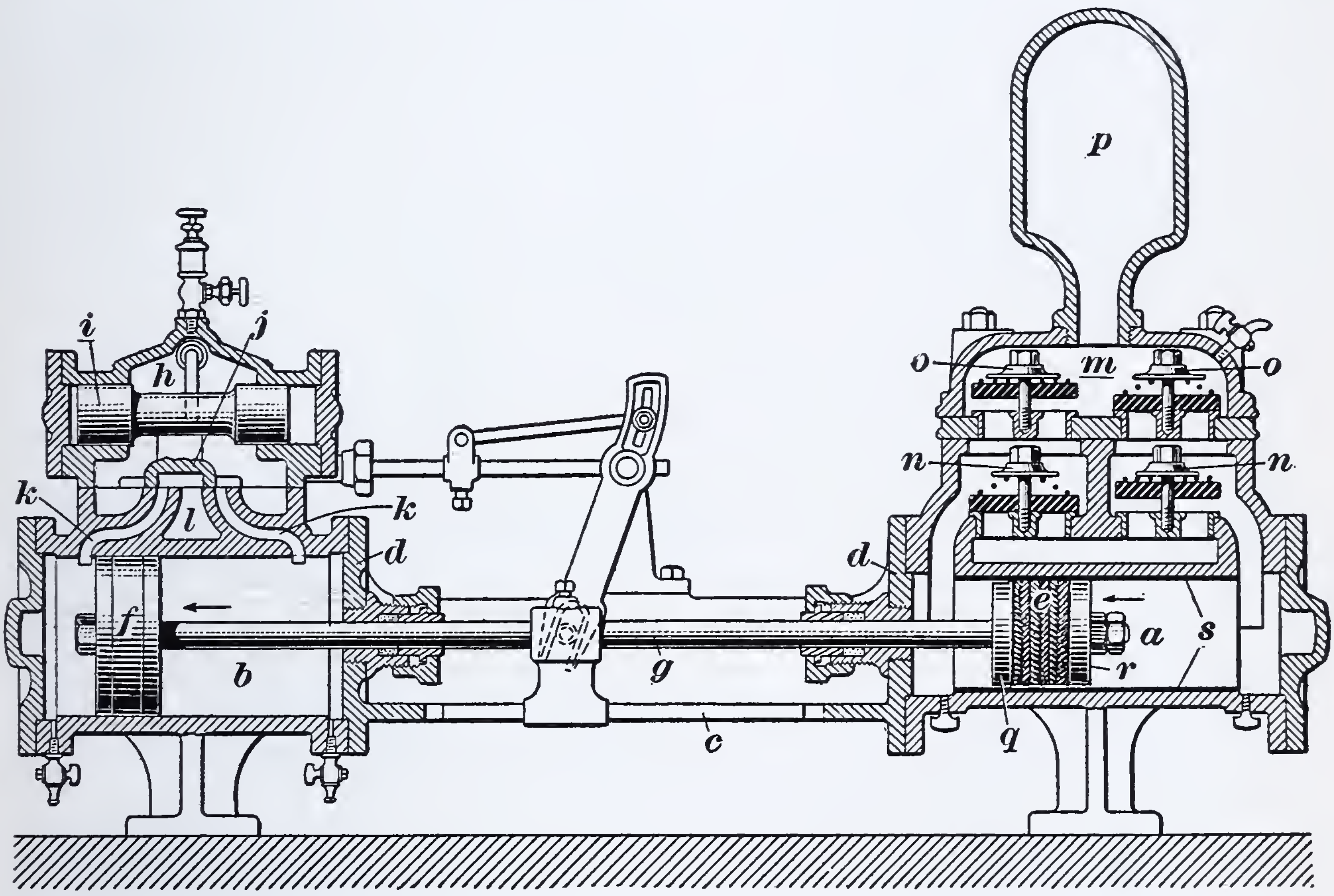


Fig. 7

water cylinder. Sometimes, the air chamber is attached to the delivery pipe close to the valve chamber as in Fig. 6.

The piston consists of a piston head q and a follower plate r, between which are held fibrous packing, leather cup rings, or metallic packing rings that fit the bore of the cylinder and prevent leakage. Fiber-packed pistons are generally used for pressures not exceeding 150 pounds per square inch. The suction valves are sometimes located above the level of the piston, as shown in Figs. 6 and 7, so that the cylinder will remain full of water and thus always be primed. Even if the valve should leak, the required amount of vacuum for suction is easily established by a few strokes of the pump.

The water cylinder should have a removable bronze lining *s* that can be replaced when worn. The valve chamber, which contains the suction valves *n* should be made with only sufficient room for the valves and port passages. In order to inspect and repair the valves and valve seats, it is necessary to have either a removable top or an opening in the top of the chamber. In the latter case, a suitable cover is also required. Generally, there are suction and discharge openings on both sides of the cylinder. The suction valves *n* and delivery valves *o* are of the disk pattern, being made of rubber or of bronze and held on their seats by cylindrical or conical springs.

21. Double-Plunger Pump.—Double-acting plunger pumps are generally used where there is a heavy pressure of

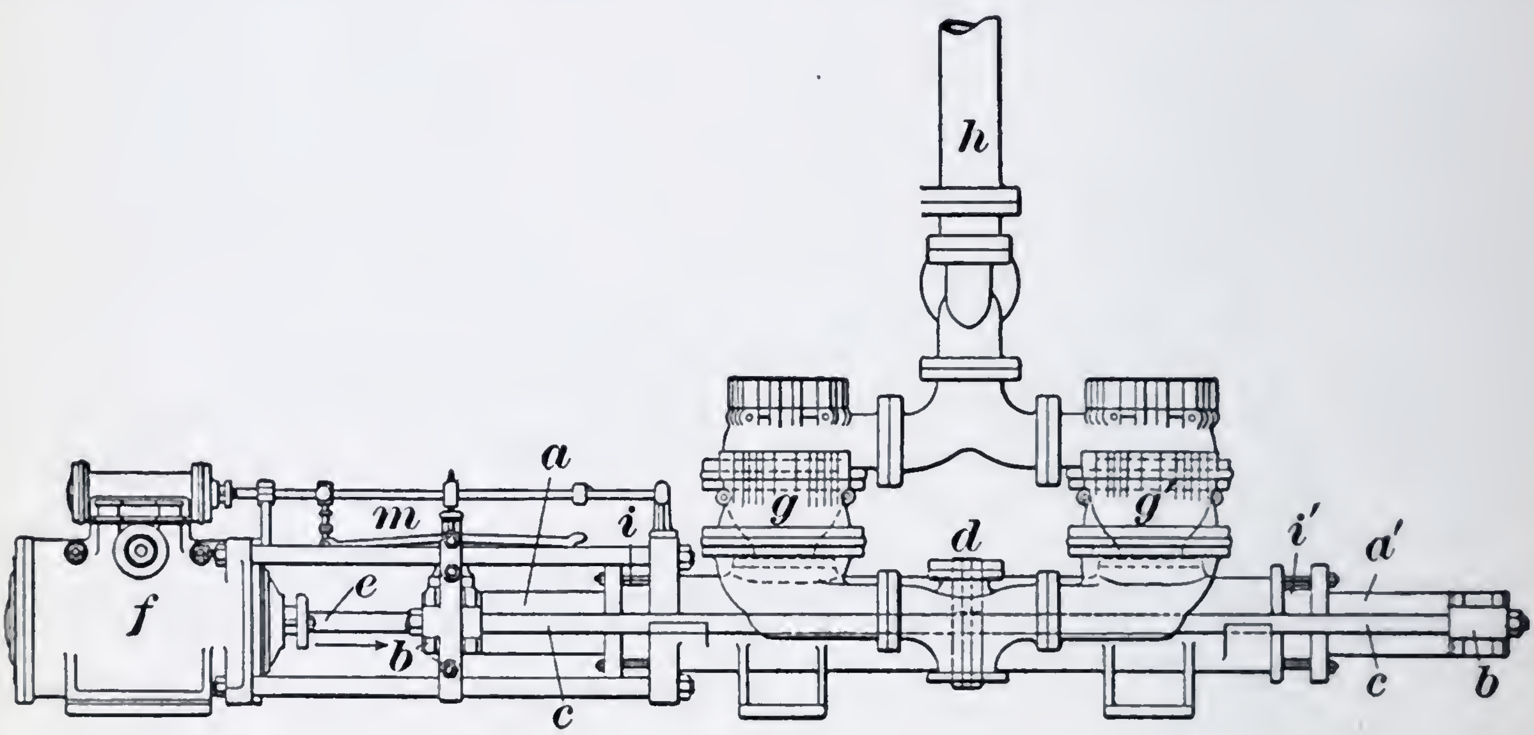


Fig. 8

water in the column pipe. A pump of this class of the double-plunger type is shown in Fig. 8. The two plungers *a* and *a'* carry yokes *b* at their outer ends and are tied together by side rods *c*. The cylinders in which the plungers *a* and *a'* work are divided by a water-tight partition *d*. The plunger *a* is attached directly to the piston rod *e*. When the steam piston in the cylinder *f* moves to the right, the plunger *a* forces water into the chamber *g* and up the discharge pipe *h*. Since the plunger *a'* is moving out of its water cylinder at the same time, water flows in through the suction pipe; and when the pump makes its return stroke the plunger *a'* forces the water

into chamber g' while the water is drawn into the cylinder in which the plunger *a* works. Thus, by the use of two plungers connected as shown, the pump is made double acting. Stuffing-boxes *i* and *i'* are used for packing the plungers, and as the plungers may be packed from outside the pump cylinder, this form of pump is called an *outside end packed pump.*

22. Single-Plunger Pump.—The Cameron pump, Fig. 9, has only one plunger and the pump is made double acting by having two pump barrels *a* and *b* placed in line with each other, and having at their ends the valve chests *c* and *d,* each of which contains a set of suction valves at the bottom and of discharge valves at the top of the chest. The suction pipe is connected

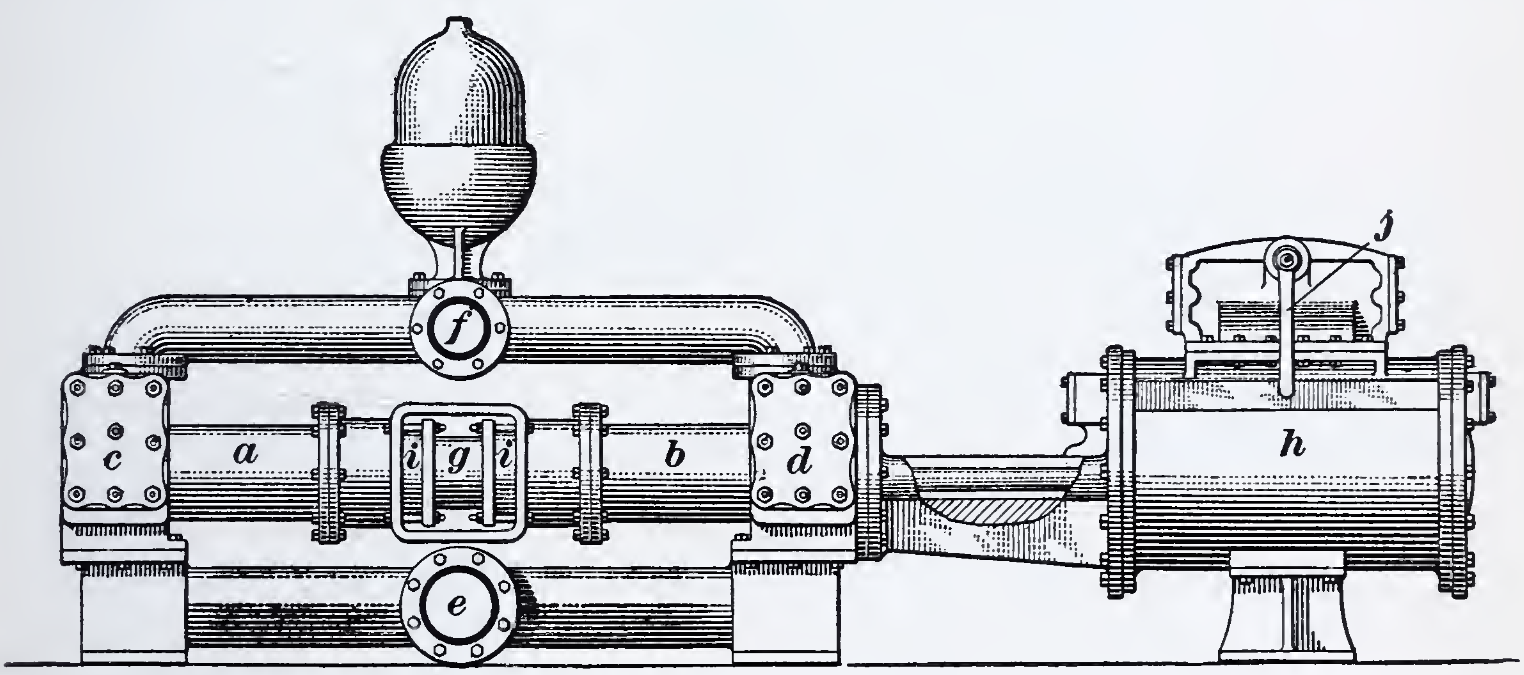


Fig. 9

at *e* and the discharge pipe at *f.* The plunger *g* works in both pump barrels, and is attached to a piston rod carrying a piston within the steam cylinder *h.* The pump is made to give a discharge equal to that of a double-acting pump by combining two single-acting pumps, the barrel *a,* valve chest *c,* and plunger *g* forming the one pump, and the barrel *b,* valve chest *d,* and plunger *g* the other pump. The plunger is made water-tight by stuffingboxes *i,* containing packing. A lever *j* allows the steam valve to be operated by hand when starting the pump.

23. Fig. 10 shows the steam end of a Cameron pump; *a* is the seam cylinder; *c,* the piston; *d,* the piston rod; *l,* the steam chest; *f,* the hollow steam-chest piston, the right-hand

end of which is shown in section; *g*, the main slide valve; *h*, the starting bar connected with a handle on the outside; *i*, the reversing valves; *k*, the bonnets over the reversing-valve chambers; and *e*, exhaust ports leading from the ends of the steam chest direct to the main exhaust which are closed by the reversing valves *i*.

The action of this valve is as follows: The interior of the steam-chest piston is, at all times, in free communication with the live-steam space of the steam chest *l*. The spaces at the ends of the steam-chest piston communicate with the live-steam space inside by small holes in the ends of the piston, one of which is shown in the right-hand section. The spaces back of the valves *i* are at all times in communication with live steam from the steam chest by means of special ports (not shown), so that the pressure keeps them in position to close the ports *e*, except at the ends of the stroke of the piston *c*, when they are pushed back so as to hold these ports open. The spaces in front of the piston valves *i* communicate with the exhaust. In the position shown, the space in the main cylinder to the right of the piston *c* is in communication with the live-steam space in the steam chest; *c* is therefore moving to the left. When *c* strikes the stem attached to the valve *i*, forces *i* to the left and uncovers the left-hand port *e*, thus allowing the steam at the left of *f* to pass out through the exhaust. The steam to the right of *f* then expands and drives *f*, and with it the main valve *g*, to the left, thus reversing the action of the steam on the piston *c*, which immediately begins to move back toward the right. Since live steam is always acting on the piston *i*, as soon as the piston *c* moves to the right, this steam pushes the piston *i* back and

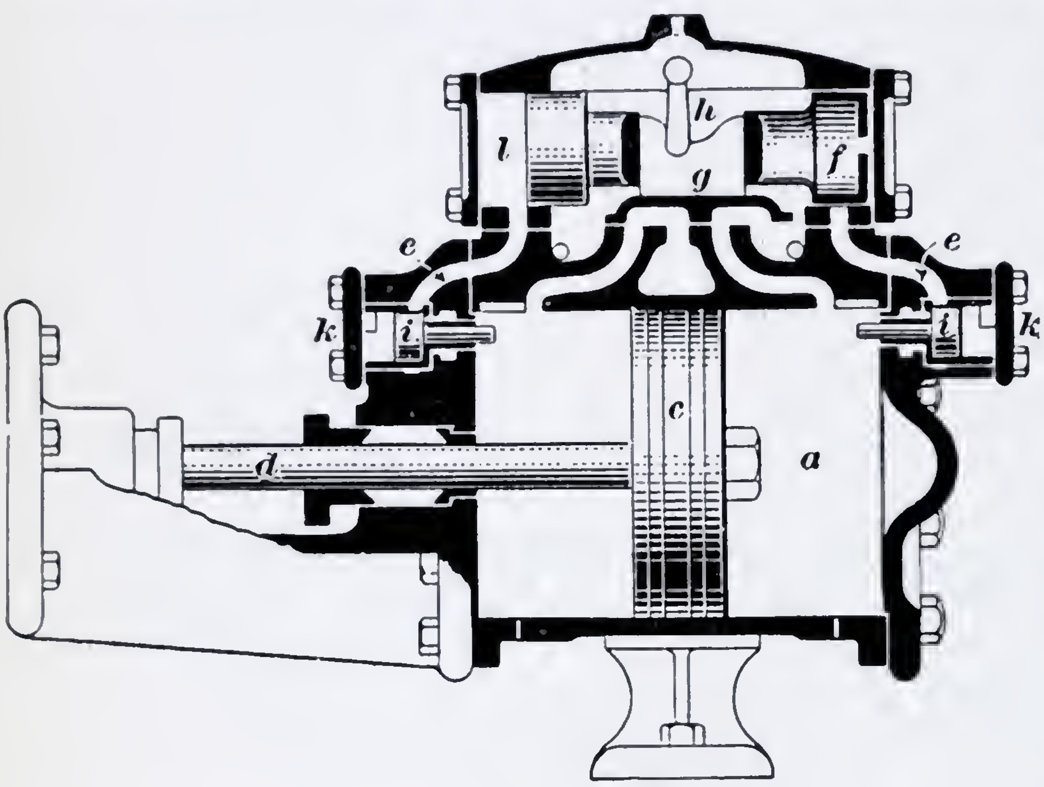


Fig. 10

covers the left port e again, after which live steam fills the port and the space connecting with it through the small hole in the end of the piston f. When the piston c strikes the stem of the right-hand valve i, the main valve is again shifted to the right and piston c is started on its stroke to the left. Steam is confined at the ends of the cylinder to prevent the piston from striking the heads.

24. Knowles Pump.—The steam end of a Knowles pump with the arrangement of the valve gear is shown in Fig. 11.

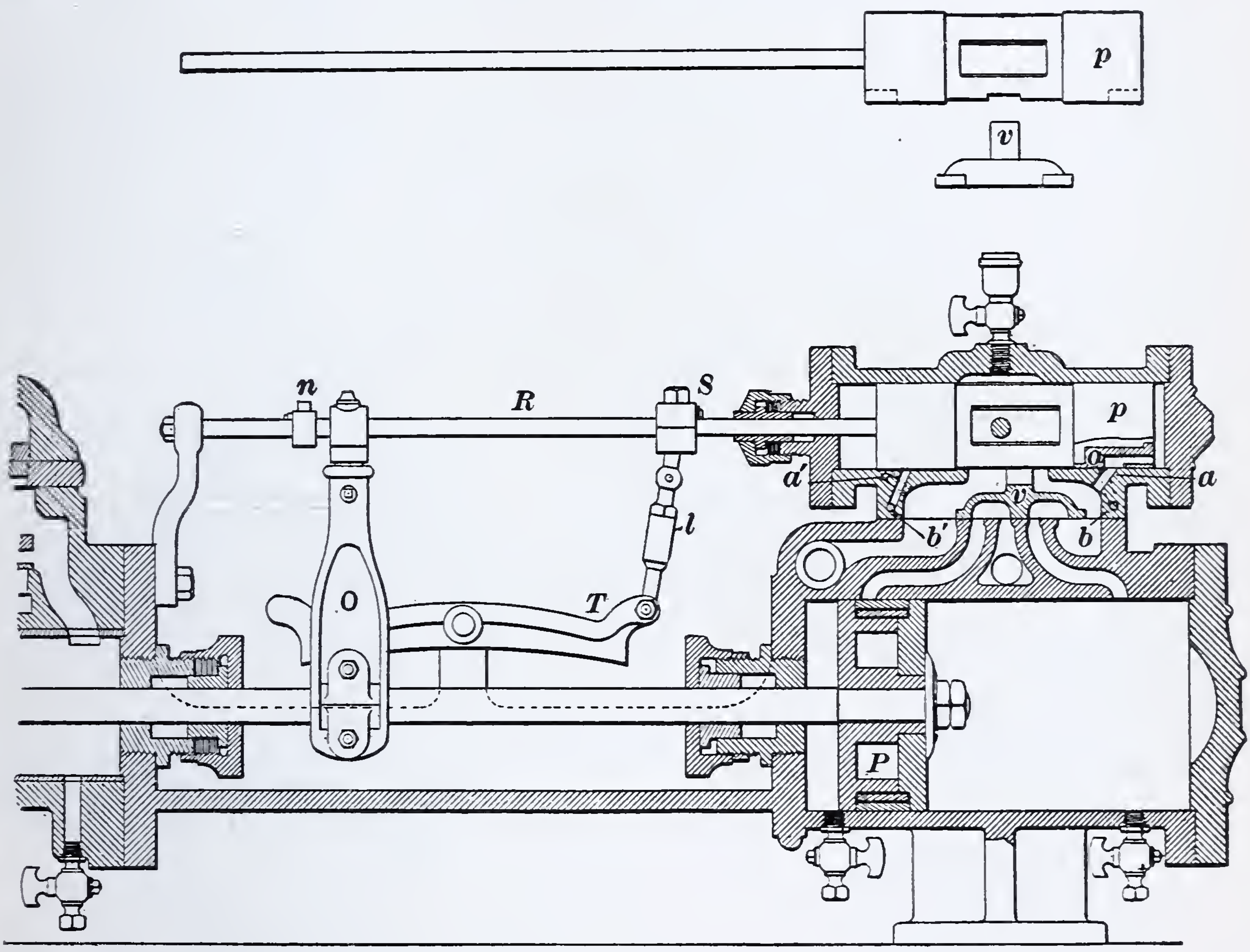


Fig. 11

An auxiliary piston p works in the steam chest and drives the main valve v. This auxiliary, or *chest piston,* as it is called, is driven backwards and forwards by the pressure of the steam, carrying with it the main valve which, in turn, gives steam to the steam piston P and operates the pump. The main valve v is a plain slide valve of the **B** form working on a flat seat. The chest piston has a rod R to which is clamped an arm S, which is connected to the rocker bar T by a link. The main piston rod carries an arm O, which is provided with a stud, or bolt, on which there is a friction roller. This roller

moves back and forth under the curved rocker bar with the motion of the main piston rod and lifts the ends of the bar, thus giving the chest piston a slight rotary motion just at the end of the stroke of the main piston. Each end of the chest piston is provided with a port *o*, shown in the right-hand end by the partial section, and the solid part of the steam chest has four ports *a*, *b* and *a′*, *b′*, which open into the space in which the chest piston moves. The ports *a* and *a′* connect with the live-steam space in the steam chest and serve as steam ports, while *b* and *b′* connect with the exhaust.

25. In the position shown in Fig. 11, the main piston has just reached that point of its stroke where the roller has acted on the rocker bar to rotate the chest piston. This has brought the port *o* (in the right-hand end of the chest piston) into communication with the live steam, admitting the latter to the space at the right of the chest piston. This steam drives the chest piston to the left and it carries the main valve *v* with it, thus exhausting the steam from the right of the main piston and admitting live steam to the left. When the main piston, under the action of this steam, approaches the right end of the cylinder, the roller lifts the right end of the rocker bar, thus rotating the chest piston so as to bring the port *o* in connection with the exhaust port *b* and the port in the opposite end of the chest piston in connection with the steam port *a′*. This drives the chest piston and main valve to the right, allows the steam at the left of the main piston to exhaust, and admits live steam to the right of the main piston again. The chest piston, as it approaches either end of its chamber, covers the exhaust port at that end, thus confining enough of the exhaust steam to form a cushion to prevent it from striking the end of the steam chest. The main piston also covers the exhaust port before reaching the end of its stroke, as shown in the figure, so that it is cushioned by the exhaust and prevented from striking the cylinder head. Special passages are provided for admitting the steam required to move the piston far enough to uncover the main ports on the return stroke. The arm *O* carries a collar that slides over the chest piston rod, and in

case the steam pressure is not sufficient to move the chest piston, this collar will strike collars, as *n*, and thus move the valve. (One of these collars is just behind the arm *S*.)

26. Duplex Pumps.—As previously stated, a duplex pump consists of two similar pumps placed side by side and so connected that the operation of one has a definite relation to the operation of the other. The two are designed to work

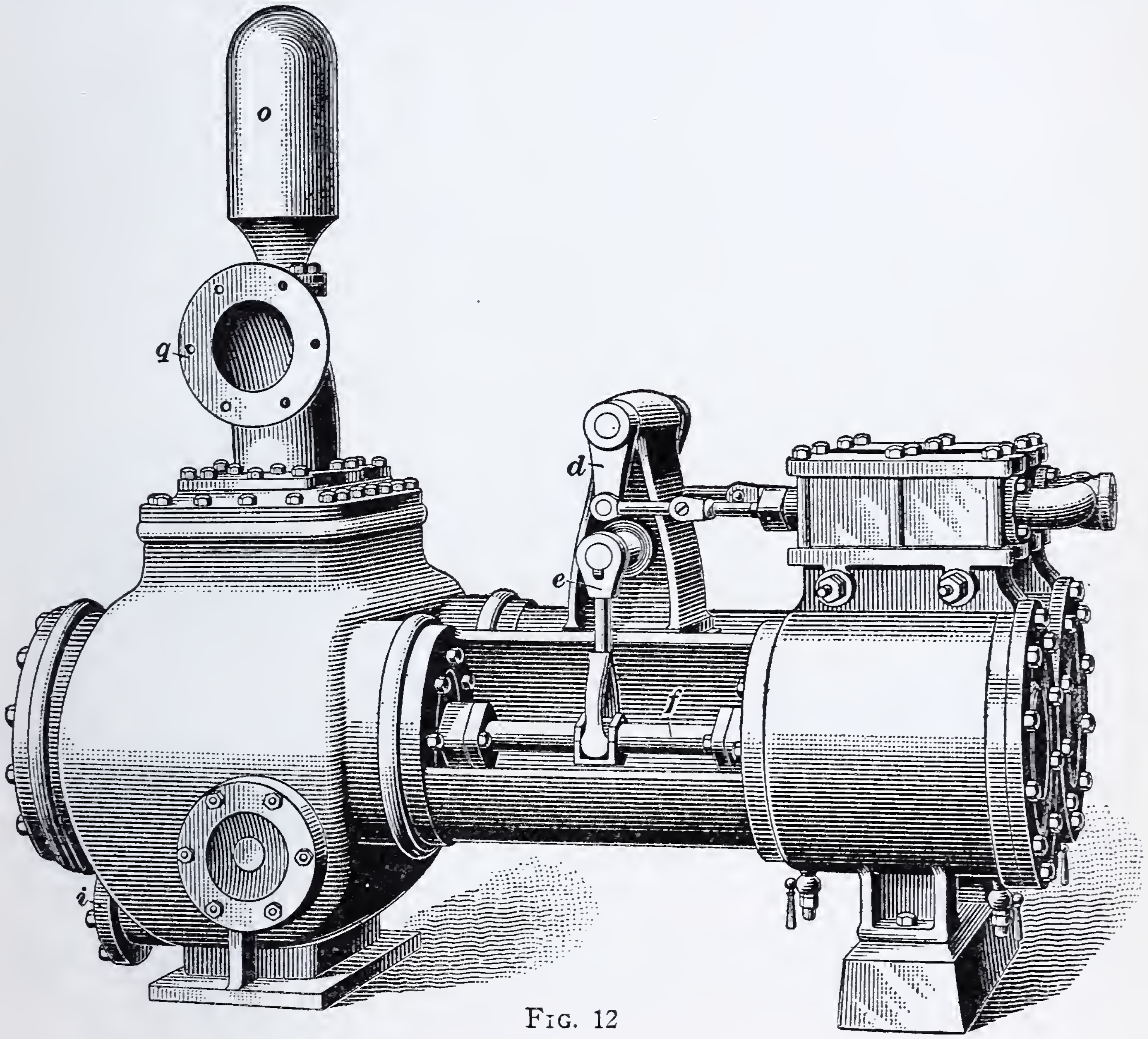


Fig. 12

together, and in most duplex types it is difficult to run one pump without the other. Duplex pumps, like single pumps, are made either with pistons or with plungers. Pistons are preferred for moderate pressures and plungers for high pressures, and, usually, each pump is double-acting.

The two single pumps that make up a duplex pump are of similar construction. A lever attached to the piston rod of one pump operates the slide valve of the steam cylinder of the other. The effect of this arrangement is such that, when the

piston or plunger of one pump arrives at a point between the middle and the end of its stroke, the location depending on the construction, the plunger or piston of the other begins its stroke, thus alternately taking up the load, producing a steady flow of water, and avoiding the stresses induced by a column of water when suddenly stopped or set in motion.

27. In Fig. 12 is shown the general arrangement of one design of a horizontal duplex pump, and in Fig. 13, a section

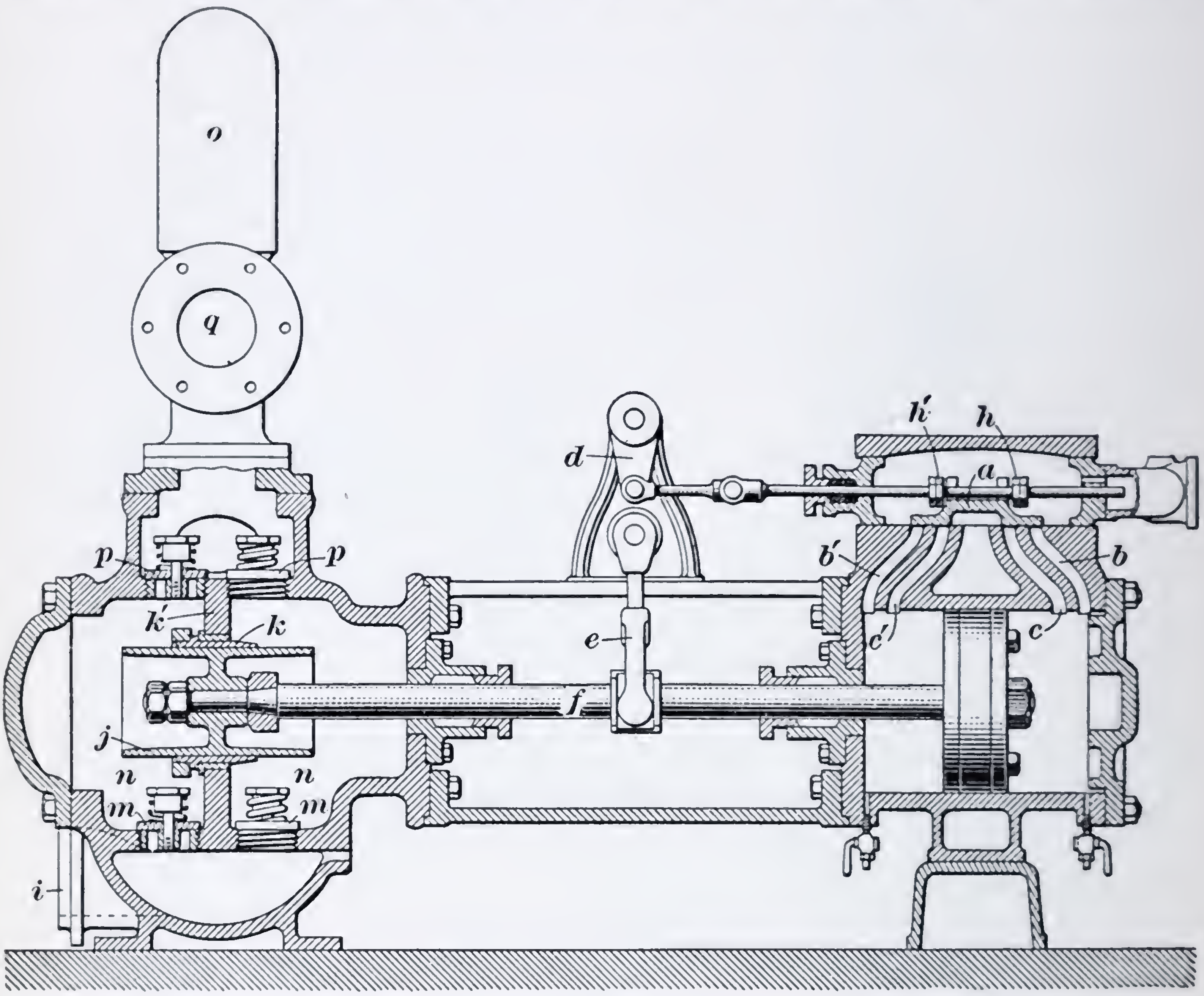


Fig. 13

through one side. The same parts have been given the same reference letters in both illustrations.

This type of duplex pump is built by many builders, and is generally known as the *Worthington duplex pump,* after the name of its inventor. The steam valves *a,* Fig. 13, of this pump are of the plain **D** type, with double ports *b, b'* and *c, c'* in the valve seat. The valve *a* of the front pump is operated by the rocker-arm *d,* which engages with the piston rod of the rear pump, while the slide valve of the rear pump is operated

by the rocker-arm *e* attached to the front piston rod *f*. The motion of one piston causes the steam to be admitted to the other cylinder, after which the piston completes its stroke and rests until its valve is moved by the other piston, when it starts on the reverse stroke. This pause of the piston allows the water valve to seat quietly. The pump will always reverse its direction of motion at the end of the stroke, as one or the other of the valves is always open and the pump is ready to start when steam is admitted. While in operation, one piston is always in motion, and at the beginning and the end of each stroke, where the motion of both pistons is the slowest, both are moving. The steam enters the cylinder through the outer ports *b* and *b′* and exhausts through the inner ports *c* and *c′*. The piston covers the inner port before it reaches the end of its stroke, and part of the exhaust steam is retained and cushions the piston, bringing it gradually to rest and preventing it from striking the cylinder head.

The steam valves have little or no outside lap, and steam is not used expansively. It is customary to allow a certain amount of lost motion between the valve stem and the valve *a*; first, to avoid getting both valves on their centers at the same time, and second, to secure a more satisfactory timing of the pistons. A simple method of providing this lost motion is to cast two vertical projections on the back of the valve *a* that will engage with nuts *h*, *h′* on the valve stem. These nuts may be so adjusted as to provide the exact amount of lost motion required between the valves and the valve stem. They also provide for adjustment of the valve on its seat.

28. The more reversals a steam pump makes in a certain time to do a given amount of work, the more wasteful it will be, because at each reversal the clearance spaces must be filled with live steam. Hence, it is important to give the pump a long stroke. The water inlet *i*, Figs. 12 and 13, is usually located so as to bring the water up between the two pump cylinders.

The water plunger *j*, Fig. 13, is double-acting and works through a deep metallic ring *k* that is bored to fit the plunger

accurately and bolted to the cylinder partition k'. The plunger is located above the suction valves m a suitable distance so as to form chambers n into which any foreign substances may fall clear of the wearing surfaces. The air chamber o is in direct line with the flow of water from the discharge valves p. When the water pressure is not higher than 150 to 200 pounds per square inch, either a plunger j, which works in a ring k, as shown, or a piston packed with fibrous packing, may be used in the water end. But if the pressure is higher, say more than 300 or 400 pounds per square inch, packed plungers are required in order to avoid excessive leakage.

29. A duplex plunger pump with two plungers A and B and two steam chests E and D is shown in Fig. 14. The

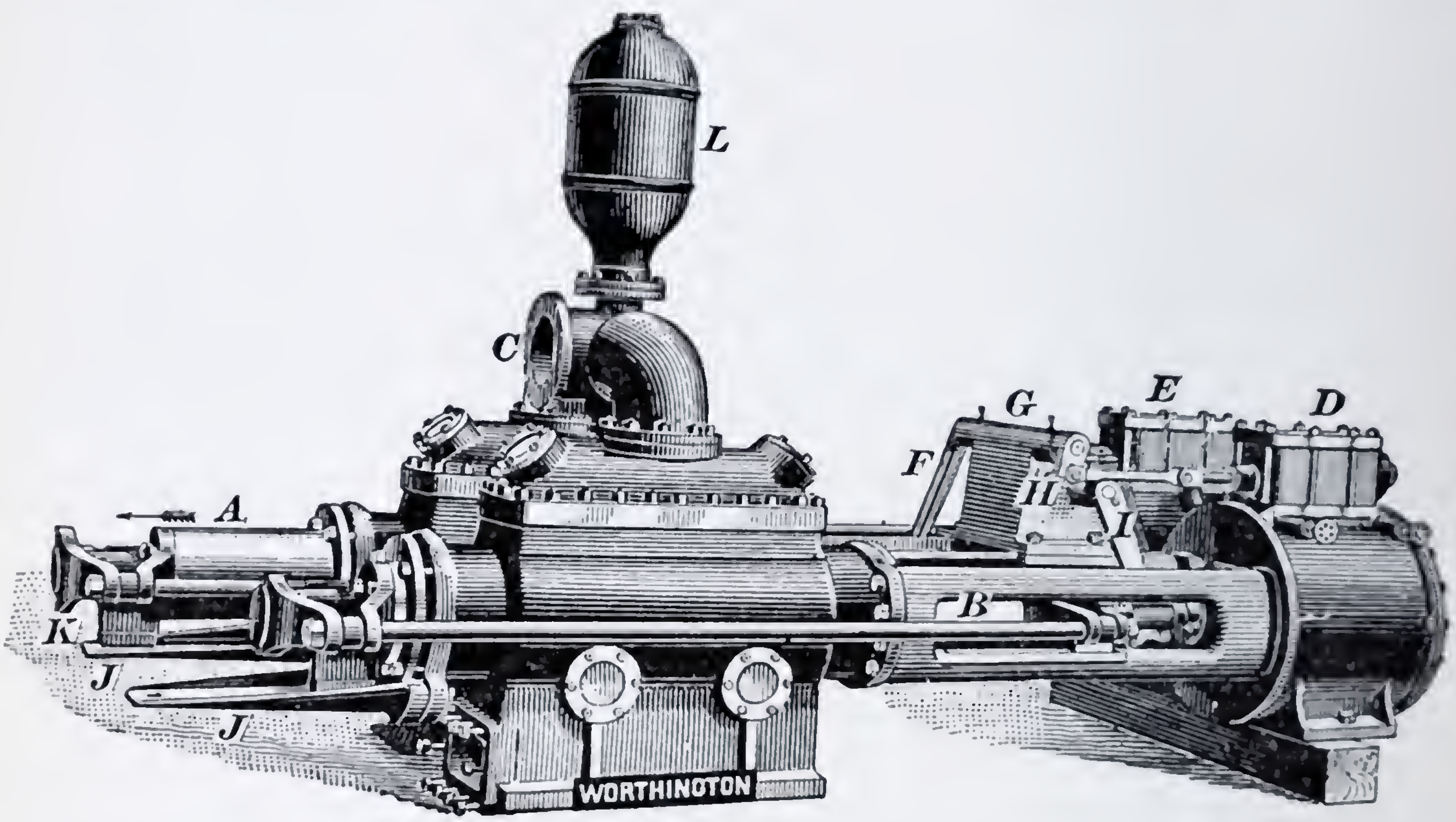


Fig. 14

plungers are each double acting, thus producing a nearly steady stream of water. The valve in the steam chest D is operated by a crank F attached to a shaft G that, in turn, moves the crank H attached to the valve stem. The valve in the steam chest E is moved in a similar manner by the crank I attached to the piston rod of plunger B. The valves are so set that the plungers A and B are always moving in opposite directions, thus giving the valves a reciprocating motion. J are slides on which the shoes K attached to the ends of the plungers move. L is the air chamber.

COMPOUND PUMPS

30. Reasons for Compounding.—In a direct-acting simple steam pump without a flywheel, the steam is not used economically, as it is admitted to the cylinder during the full length of the stroke, and is not therefore used expansively. Although simple direct-acting pumps are usually low in first cost and simple in construction, they are not economical of fuel. Therefore, they are not adapted for large underground pumping installations, when fuel economy is an essential; for, in addition to the uneconomical use of steam in such an engine, there is always a considerable loss in transmission through condensation in long pipe lines. To overcome this difficulty, compound triple-expansion or quadruple-expansion engines are

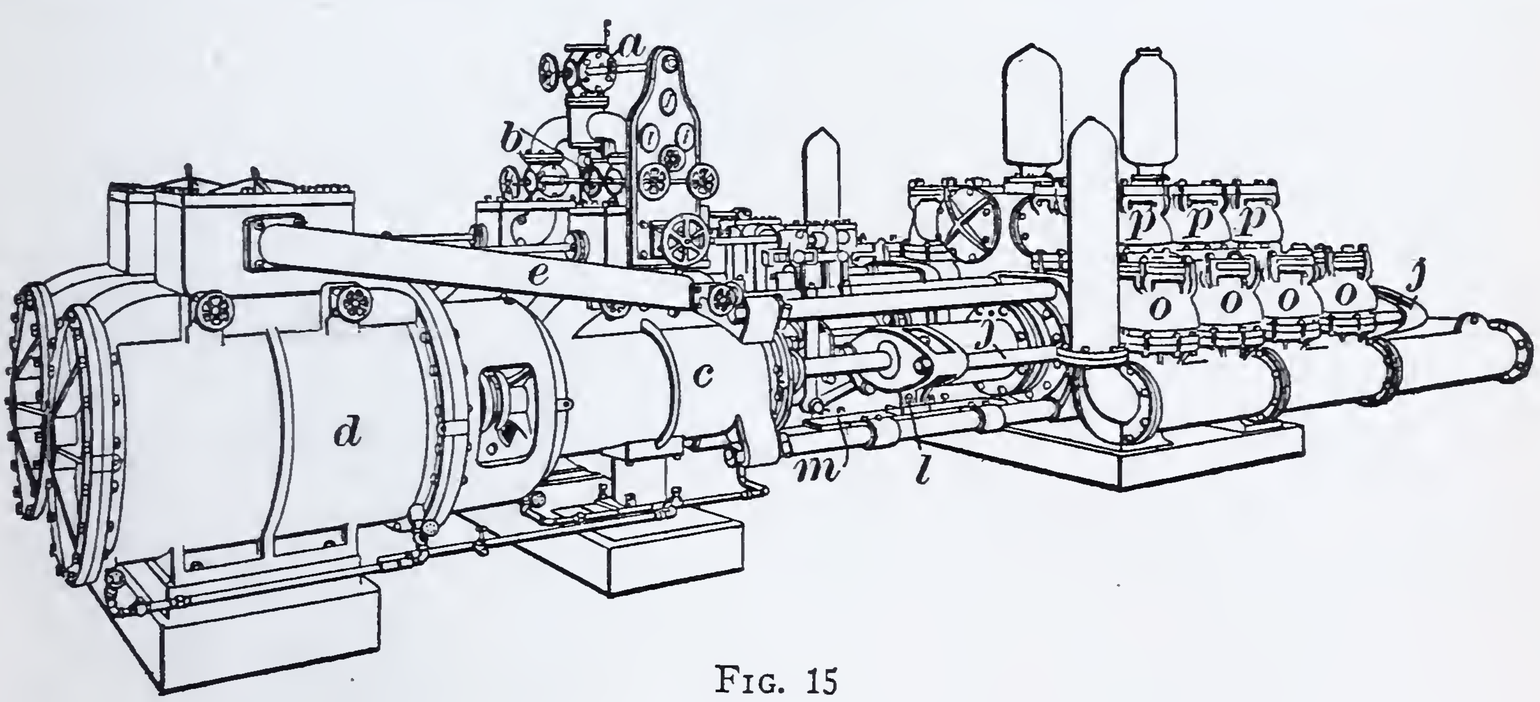


Fig. 15

used quite extensively in connection with large pumping installations, particularly in localities where great attention must be paid to fuel economy, as, for instance, in the Lake Superior region.

Compound pumps may be made either single or duplex, and the water ends may be either of the piston or plunger type. The cylinders, in compounding, are usually arranged tandem and applied to direct-acting pumps, although cross-compound engines are sometimes used.

31. A duplex, tandem, compound, outside-packed, mine pump is shown in perspective in Fig. 15 and in section in Fig. 16. For a rated capacity of 2,000 gallons of water per

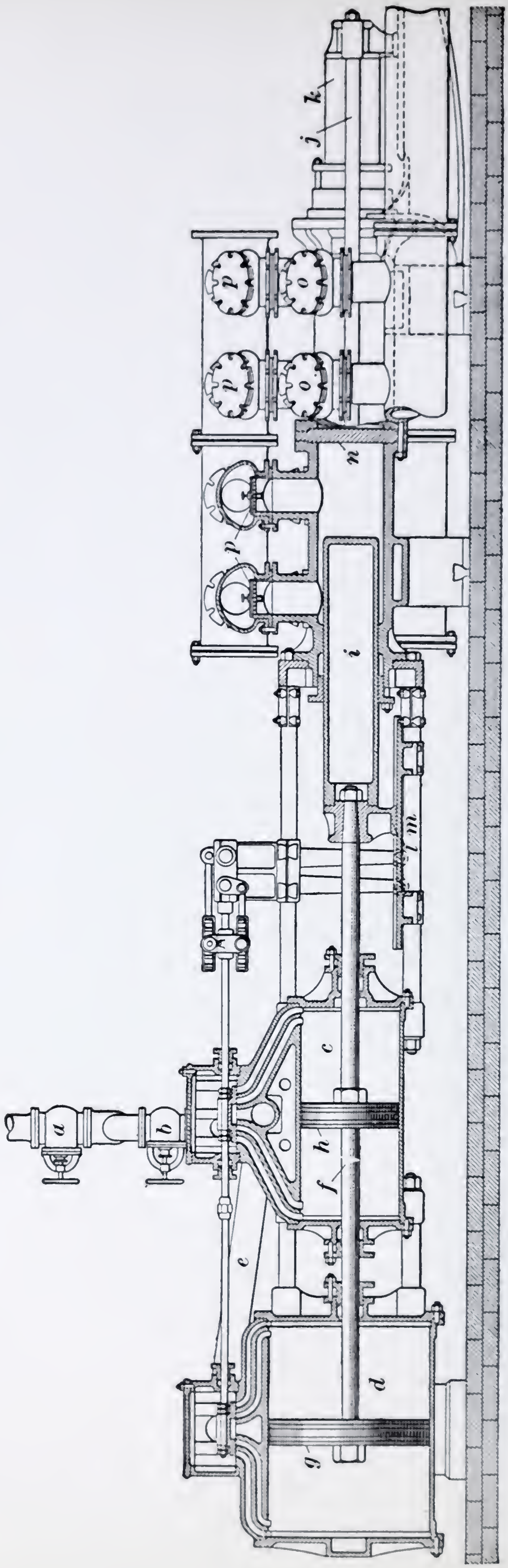


FIG. 16

minute against a head of 425 feet, the principal dimensions of this pump are as follows: diameter of high-pressure cylinder, 25 inches; diameter of low-pressure cylinder, 42 inches; diameter of pump plunger, 14 inches; stroke of plungers and pistons, 48 inches.

In the two illustrations, *a* is the main throttle valve and at *b* are auxiliary throttle valves in the pipe leading from the main valve to each of the high-pressure cylinders *c*. The object of these auxiliary valves is to permit the amount of steam entering each side of the pump to be regulated so that each side may do an equal amount of work, as better results are obtained when a pump is working in this way. The exhaust passes directly from the high-pressure cylinder *c* to the low-pressure cylinder *d*; and since there is no cut-off in either cylinder, the back pressure on the high-pressure piston is equal to the pressure on the low-pressure piston, except for the resistance to the flow of steam through the ports and the pipe *e*. The piston rod *f* is direct-connected to both the high- and the low-pressure steam pistons *g* and *h* and to the water plunger *i*; and by means of a yoke *j* to another plunger at *k*. The end of each plunger is supported on shoes *l* that run on slides *m*. A partition, or diaphragm, *n* separates the two plunger cylinders. The steam valves are of the ordinary **D** type. There are four valves in each plunger, two suction valves *o* and two discharge valves *p*, making a total of sixteen valves for the whole pump. For a pump of this size, it is often customary to have a large number of small valves, from fifty to one hundred, instead of a small number of large valves as here shown; but if the pump is handling acid water, which attacks not only iron but brass and phosphor-bronze as well, the life of the valve seat is very short, and the valves must therefore be made strong and so that they can be easily examined and quickly replaced. The parts of a large valve, being heavier, are not liable to be twisted or broken by rough usage, and it is, as a rule, less work to care for a few large valves than for a great number of small ones. A pump of this kind is therefore especially adapted for pumping under conditions where the valves must be frequently repaired and replaced.

32. Since the length of the stroke is the same for both cylinders, the number of expansions in a compound pump of the type just shown is equal to the ratio between the areas of the low- and the high-pressure pistons, the space of the valve passages and of the connecting pipe being neglected.

EXAMPLE.—What is the back pressure on the high-pressure cylinder in the pump shown in Figs. 15 and 16 if the steam is admitted to that cylinder at an absolute pressure of 114.7 pounds per square inch, the atmospheric pressure being 14.7 pounds per square inch?

SOLUTION.—The area of the high-pressure piston is, 25×25×.7854 =490.87 sq. in. The area of the low-pressure piston is, 42×42×.7854 =1,385.44 sq. in. If the absolute pressure is 114.7 lb. per sq. in., the

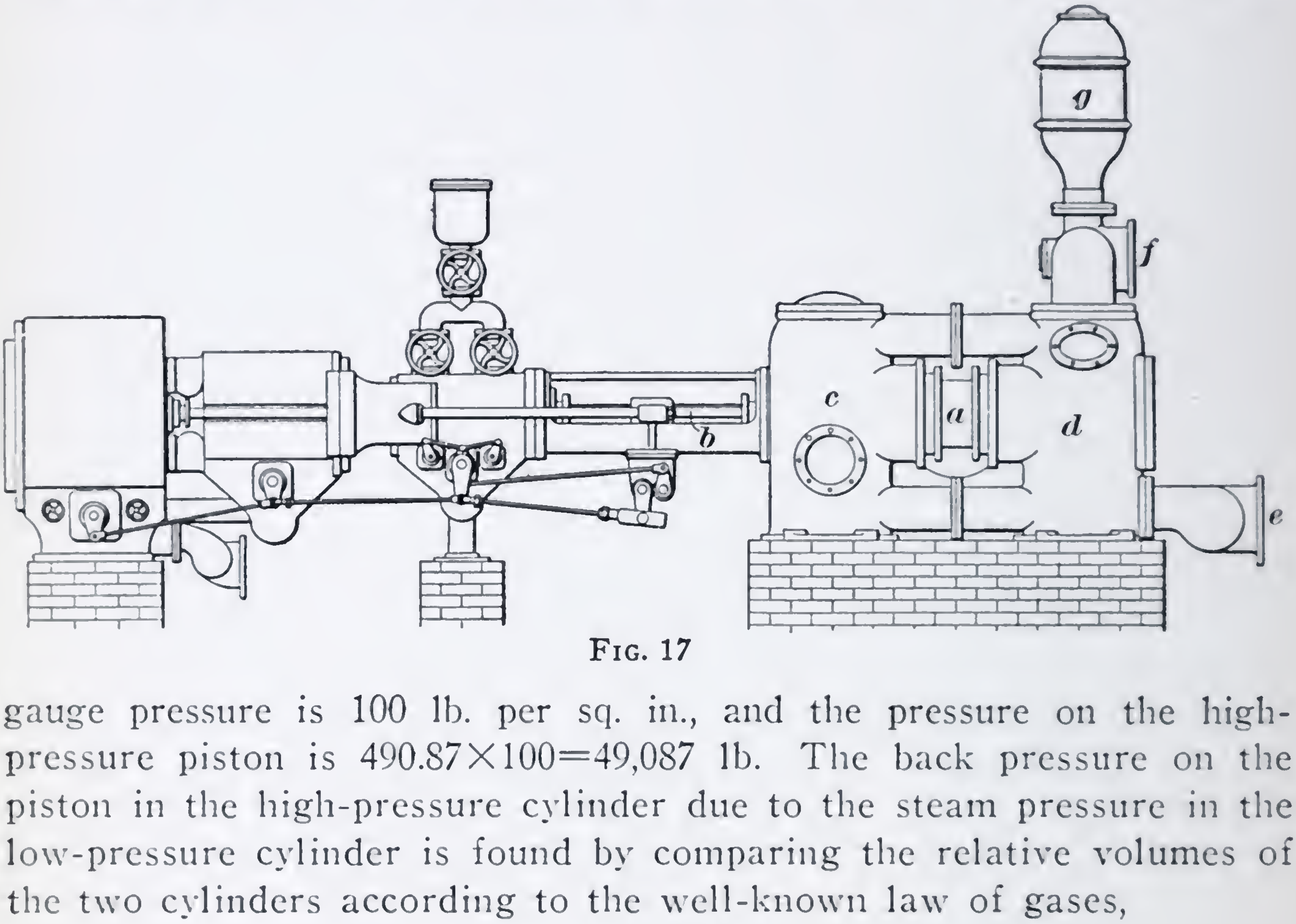


FIG. 17

gauge pressure is 100 lb. per sq. in., and the pressure on the high-pressure piston is 490.87×100=49,087 lb. The back pressure on the piston in the high-pressure cylinder due to the steam pressure in the low-pressure cylinder is found by comparing the relative volumes of the two cylinders according to the well-known law of gases,

$$p\,v = p'\,v'$$

in which p = absolute pressure, in low-pressure cylinder;

p' = absolute pressure, in high-pressure cylinder;

v = volume of low-pressure cylinder;

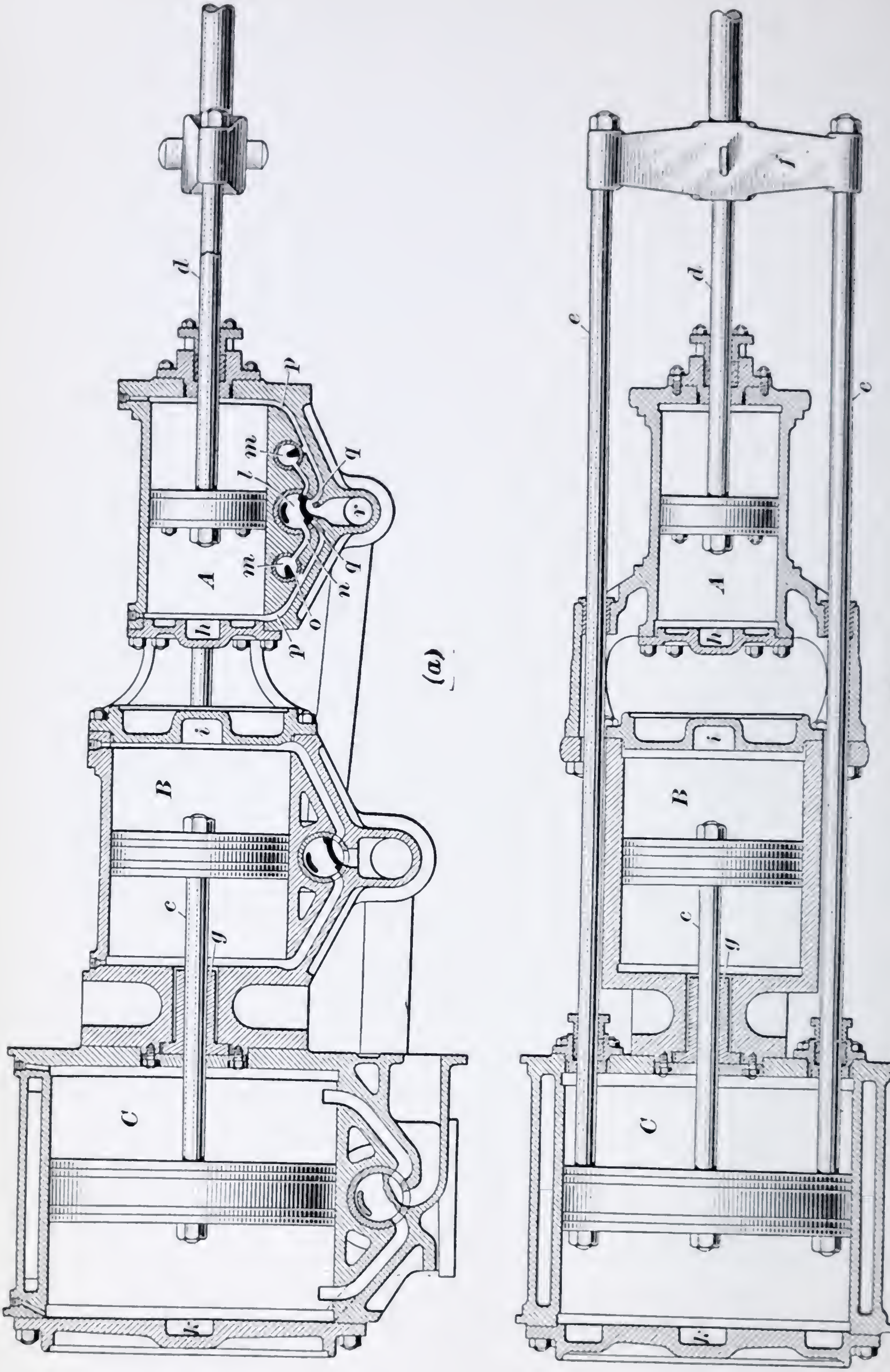
v' = volume of high-pressure cylinder.

The volumes are directly proportioned to the squares of the diameter, hence, $p = p'\frac{v'}{v} = p'\left(\frac{d'}{d}\right)^2 = 114.7 \times \frac{25^2}{42^2} = 40\ 6$ lb. $40.6 - 14.7 = 25.9$ lb. per sq. in. gauge pressure in low-pressure cylinder, and 490.87×25.9 =12,713.53 lb. back pressure on high-pressure cylinder. Ans.

33. Triple-Expansion Pumps.—One side of a double-acting, triple-expansion, duplex, Worthington mine pump having center-packed plungers is shown in Fig. 17. The water end of this pump is called the Scranton type by the manufacturers. The plunger *a* is connected to the piston rod *b* and works in the pump chambers *c* and *d*, which have suction valves at the bottom and delivery valves at the top. The suction pipe is connected at *e* and the discharge pipe at *f*. An air chamber *g* above the discharge valves prevents shocks and provides a continuous discharge.

The arrangement of the steam end of this pump is shown in detail in Fig. 18 (*a*) and (*b*). One object of this design is to render all of the piston accessible through the cylinder covers *h*, *i*, and *k*, and at the same time to avoid the use of a stuffingbox between the high-pressure cylinder *A* and the intermediate cylinder *B*. The low-pressure piston and the intermediate piston are connected by the piston rod *c*, and the low-pressure piston is connected to the high-pressure piston rod by side rods *e* and yoke *f*. The bushing *g* between the intermediate and low-pressure cylinders can move sidewise so as to accommodate any want of alinement between the two cylinders. At the same time, it prevents leakage of steam between the cylinders.

34. The valve *l* is the high-pressure distributing valve and at *m* are the high-pressure cut-off valves. Steam enters through the center of the valve *l* and passes through the port *n* and through the cut-off port *o* into the high-pressure cylinder by way of the port *p*. The cut-off is effected by turning the rotary valves *m*. Exhaust from the high-pressure cylinder takes place through the ports *q* and thence into the high-pressure exhaust pipe *r* that leads to the center of the intermediate valve *s*. The valve *s* is also a rotary valve designed to distribute the steam in exactly the same manner as the common **D** slide valve. The intermediate and low-pressure cylinders are not provided with cut-off valves. The exhaust steam from the intermediate cylinder passes out through the port *t* into the exhaust pipe *u* and thence to the center of the low-pressure



(a)

(b)

Fig. 18

distributing valve *v*. From the low-pressure cylinder, the steam exhausts into the chest *w* and thence to the condenser or atmosphere. Relief valves, which are not shown in the illustration, are also provided on the low-pressure cylinders only. The distributing valves are worked from the pump on the opposite side, while the cut-off valves are worked from the pump that is in line with the steam end to which they are connected.

CONDENSING PUMPS

35. Purpose of Condensers.—From the standpoint of operation, and particularly of economy, it is often advisable and frequently necessary to use a condenser in connection with a steam-driven pump. The use of a condenser results in an appreciable reduction in the amount of fuel and feedwater consumed and in the size of the boiler plant, as compared with these requirements when the pump is operated without a condenser. Such an apparatus is attached to the exhaust pipe of the pump and, by the use of water, condenses the steam and produces a vacuum on the exhaust side of the steam piston. This vacuum, of course, decreases the pressure on the exhaust side of the piston and has the same effect as an equivalent increase in pressure on the opposite or pressure side of the piston. A vacuum of 24 to 25 inches, or an absolute pressure of 6 or 5 inches, of mercury is considered good practice in mine pumping. Since the mean pressure of the atmosphere at sea level is 14.7 pounds per square inch, when steam is exhausted directly into the air there is a back pressure of that amount acting against the piston; but when a condenser is used, this back pressure is reduced to $\frac{6}{30} \times 14.7 = 2.94$, or about 3 pounds per square inch.

The use of a condenser on small pumps, used merely for local pumping, is, of course, not economical, but wherever the pumping plant is of considerable size condensers should be used. Another advantage resulting from the use of a condenser is that the steam is not exhausted into the mine air, where it causes the timbers to rot, the roof to spall, and iron to rust. Also, humid warm air is injurious to the workmen.

36. Types of Condensers.—Condensers may broadly be divided into two distinct classes: *Surface condensers,* in which the steam and cooling water are kept separate from each other by tubular condensing surfaces; and *direct-contact condensers,* wherein the steam and the cooling water come in direct contact and mingle intimately with each other. The completeness with which the contact between the steam and water can be accomplished determines to a large extent the efficiency of the apparatus. There are numerous types of both of the foregoing classes, but their primary principles are similar.

37. Surface Condensers.—A surface condenser usually consists of a rectangular shell made of cast iron or steel plate, fitted with banks of brass or copper tubes placed in inner heads near the ends of the shell. The tube ends are expanded firmly into place in one head, while the opposite ends are fitted with slip joints and packing in the other head to allow for the expansion and contraction that takes place under the variable temperatures to which such apparatus is subjected. Exhaust steam fills the shell, is condensed by contact with the water-filled tubes, drops to the bottom of the shell, and is removed by an air pump.

In *single-flow surface condensers* the cooling water enters at one end of the shell, passes directly through the tubes into the opposite head and thence out of the discharge opening to the atmosphere. In *multiflow surface condensers,* partitions are arranged in the inner heads whereby the cooling water, instead of passing directly through the tubes and out, is caused to flow through the lower set of tubes in one direction and an upper set of tubes in the opposite direction. This reversal of flow may be arranged to take place several times, thus making the multiflow principle.

38. One form of surface condenser is shown in Fig. 19. The shell *a* has two heads that hold the upper and the lower banks of tubes *b* and *c*. The pump *d* forces cooling water through the connection *e* into the lower bank *c* of tubes, whence it passes through the left-hand head to the upper bank *b* of tubes and then out at *f*. The exhaust steam from the pump

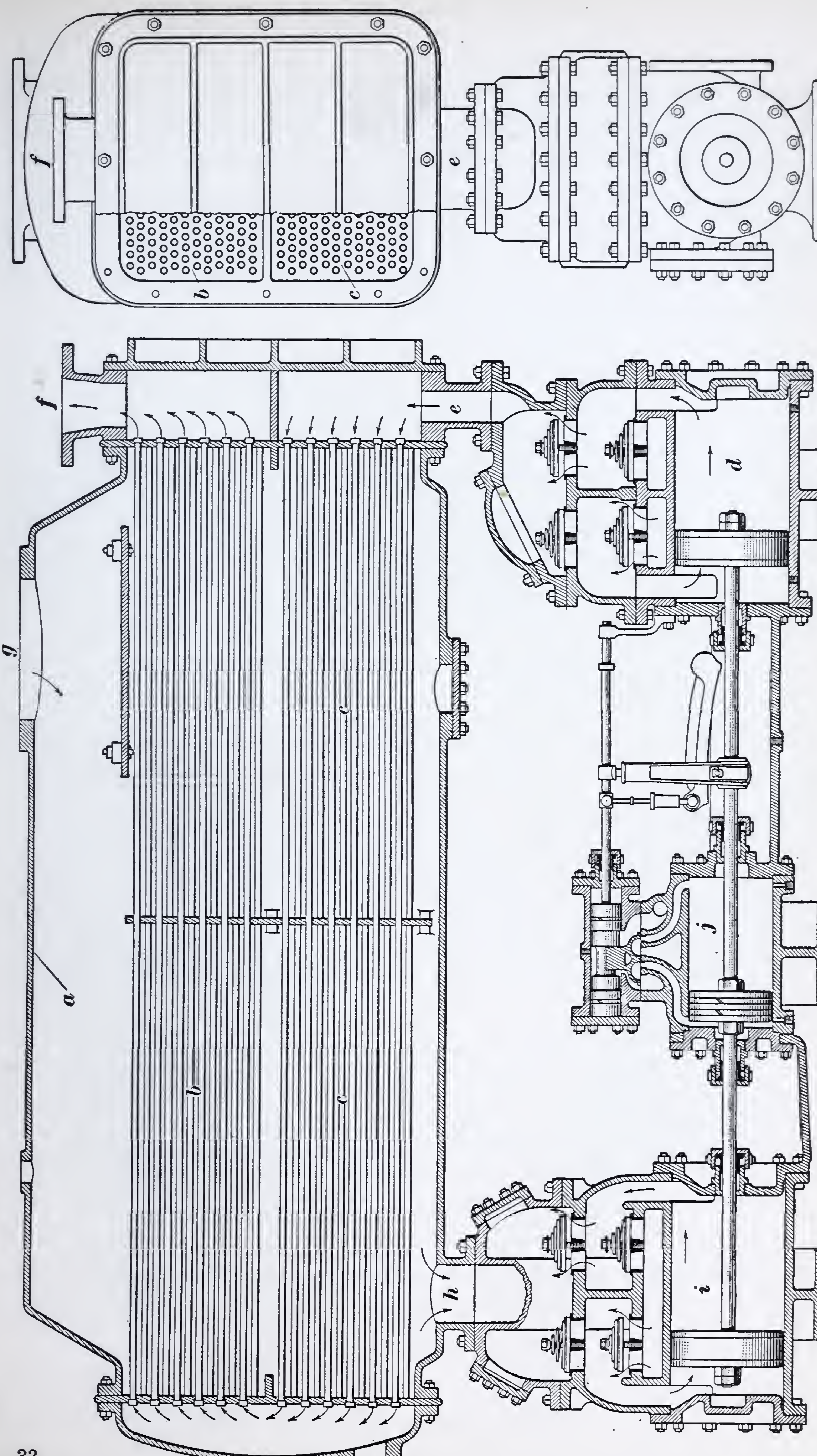


Fig. 19

enters at *g* and comes in contact with the water-cooled tubes, with the result that it is condensed. The condensate falls to the bottom of the shell *a* and flows through the connection *h* into the condensate pump *i* by which it is removed. The steam cylinder *j* drives the water pump *d* and the condensate pump *i*. The latter not only removes the condensed steam, but also the air that is carried into the condenser with the steam.

39. The surface condenser is particularly advantageous where the cooling water is unfitted for boiler feed, and when no cheap supply of boiler feed is available. The condensed steam from a surface condenser makes the best boiler feed, as it is pure distilled water, entirely free from any scale-forming matter, and containing a considerable amount of heat, as compared to water of ordinary temperature. When condensed steam is used for boiler feed, a suitable oil separator should be placed in the exhaust pipe between the pump and the condenser in order to protect the boilers from accumulations of grease and oil. Although water that may be unfit for boiler-feed service can be used for condensing purposes, acidulous water should be strictly avoided. The direct-acting pump *i*, Fig. 19, which handles both non-condensible gases and condensed steam, is the simplest arrangement for plants where moderate vacuum is desired. It is absolutely necessary to place the air pump below the condenser body in order to obtain satisfactory results.

40. Jet Condensers.—The direct-contact form of condenser embraces a number of different types classified principally by the method employed for removing the water. The most common types are the jet condenser and the barometric condenser. The jet condenser is generally a pear-shaped cast-iron vessel in which the steam is brought into direct contact with the cooling water. The combined mass falls to the bottom of the condensing chamber, from which it is removed by a pump provided for that purpose.

In Fig. 20 is shown a jet condenser, in which *a* is the condensing chamber and *b* is the air pump driven by the steam cylinder *c*. The cooling water, or injection water, enters the

condensing chamber at its top through the opening *d*, and its distribution is to some extent controlled by a spray *e*, adjust-

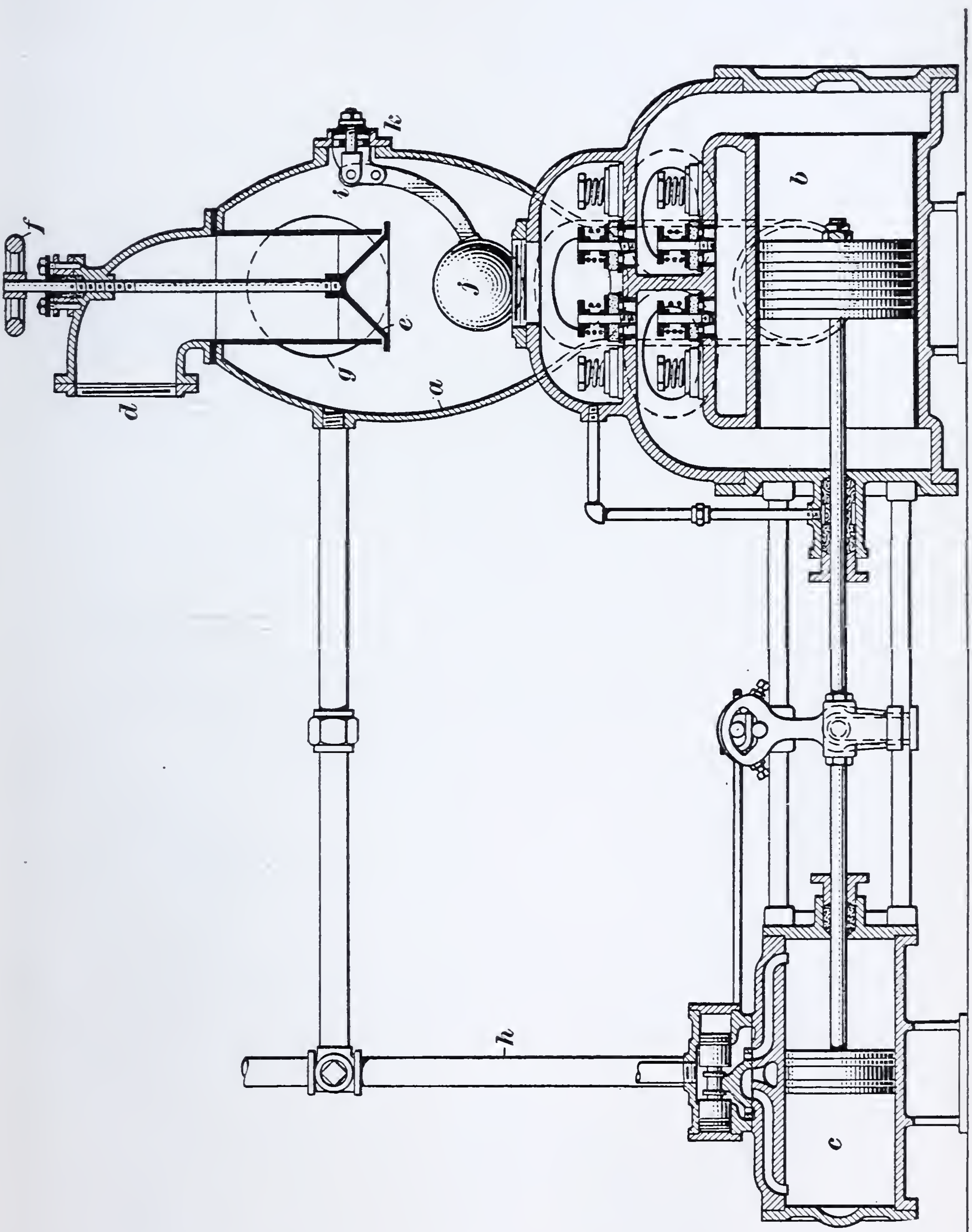


Fig. 20

able from the exterior by the valve stem to which is attached the wheel f. This spray head not only regulates the quantity

of water flowing, but also distributes it in such a manner as to bring about the intimate mixing of the entering steam and the cooling water. The exhaust steam enters the condensing chamber through an opening *g* below the point of water entrance, comes in direct contact with the water issuing from the spray head, is condensed, and together with the injection water and non-condensible gases is drawn out of the bottom of the condenser chamber by the air pump *b*. The exhaust steam from the cylinder *c* is led to the condenser chamber through the pipe *h*.

41. The cylinders of the pump whose steam is being condensed must be protected from becoming flooded through failure of the condensing apparatus to act properly. As such failure would permit an undue accumulation of water in the condenser, with consequent damage to the pump, a device called a *vacuum breaker* is used to prevent such accident. It is placed at a suitable height within the body of the chamber, as at *i*, Fig. 20, and consists of a ball float *j* that rises and opens the valve *k* and admits air to break or destroy the vacuum and stop the supply of injection water in case the condenser fails to act as it should. The float is so adjusted as to break the vacuum when the level of the water in the condensing chamber rises to a certain point. The water is thus prevented from being drawn back through the exhaust-steam pipe to the pump cylinder, where it might cause the cylinder or cylinder head to be broken. This type of condenser is best adapted for use with the ordinary medium- or high-speed reciprocating pump or engine. The amount of injection water required varies from 20 to 30 times the weight of steam to be condensed, depending upon the temperature of the water. In order to secure the most satisfactory results in operation, the vertical distance between the surface of the injection-water supply and the center of the injection-water inlet *d* should not exceed 18 feet, and the condenser should be so located that the center of the exhaust-steam inlet *g* is not less than 4 feet below the bottom line of the engine or pump cylinder. This is an added precaution against flooding the cylinders of the pump or engine.

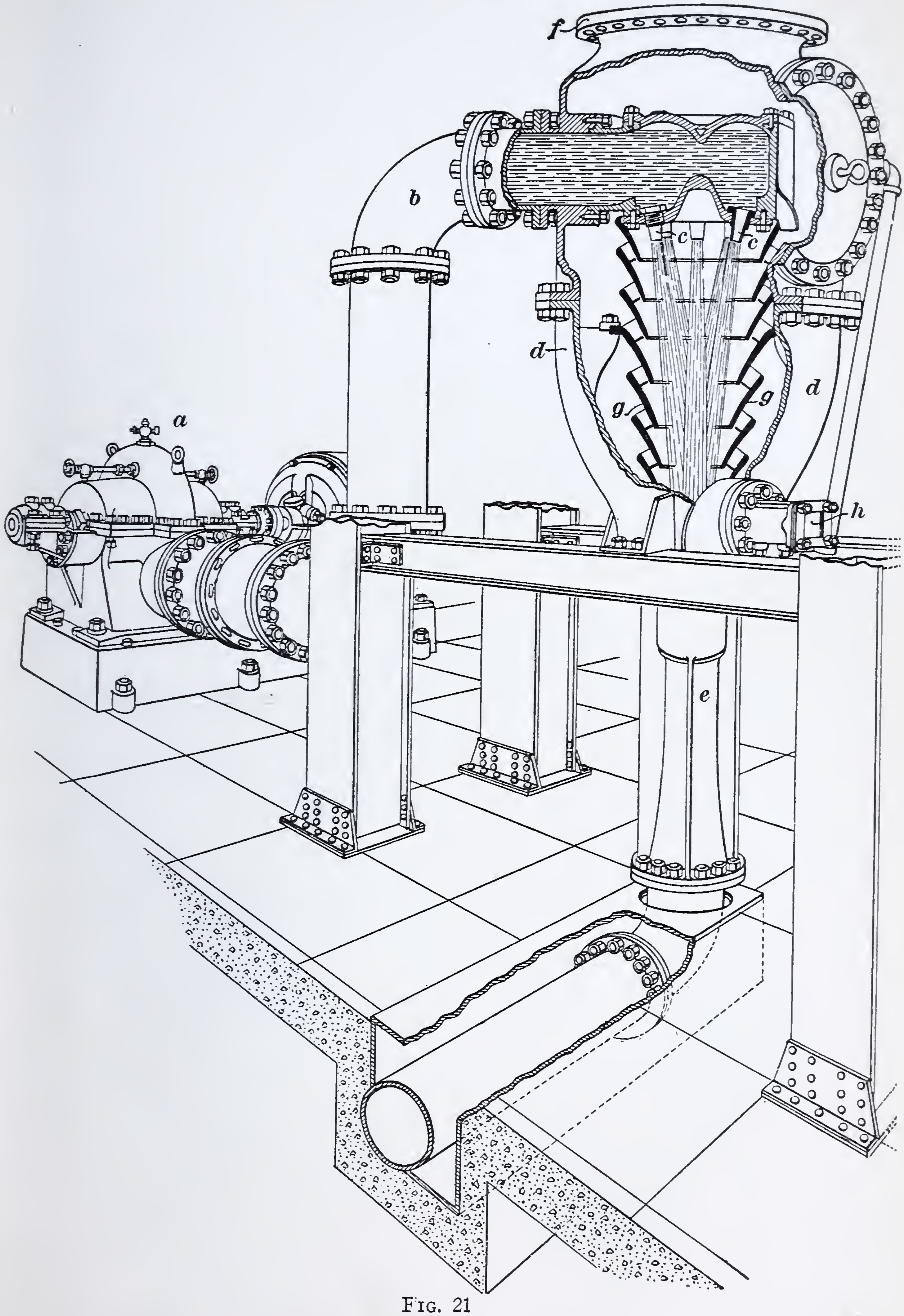


Fig. 21

42. The low-level multijet condenser is of simple design, acts with ease and satisfaction, and does not require a vacuum pump for the removal of water and contained gases. Its general arrangement is shown in Fig. 21, wherein a standard centrifugal pump *a* discharges cooling water under light pressure through the pipe *b* to a number of separate nozzles *c* inside the condensing chamber *d*. The jets from these numerous nozzles converge in the neck of a discharge pipe *e* that leads to the hot well. The exhaust steam enters the condensing chamber through a pipe connected at *f*, flows down through the openings between the cones *g*, mingles with the jets of water, and is condensed. The condensation and the cooling water then pass to the hotwell through the discharge pipe, taking with them any air that may have entered the condenser. A vacuum breaker *h* is provided to prevent damage to the pump or engine cylinder in case of failure of the condenser.

43. A striking feature of the condenser shown in Fig. 21 is that only one standard centrifugal pump is required, and no air pump, hydraulic or steam ejector connections are necessary for the removal of the air or other non-condensible gases. The pump used to supply injection water may be placed at any convenient point where its efficiency will not be unduly decreased by a suction lift exceeding 18 feet. To start this apparatus, the injection water pump is primed in the usual manner; and after it lifts the water and is brought up to speed, the water discharging through the condenser head quickly builds up the vacuum. To shut down the condenser, it is only necessary to close the injection valve on the discharge line between the pump and the condenser, when the vacuum breaker will instantly open and destroy the vacuum. Depending upon conditions of vacuum and injection water, this type of condenser requires from 25 to 75 per cent. more water than the ordinary jet condenser, but this feature is fully compensated for, so far as power required for operation is concerned, by the high efficiency of the standard injection-water pump operating under normal head as compared with a removal pump operating under high vacuum, and by the elimination of the air pump.

44. Barometric Condenser.—In the barometric condenser the condensation takes place in a chamber practically identical with the chamber of a jet condenser, but elevated so that the water falls by gravity to the atmosphere. One pound of steam at atmospheric pressure occupies 1,642 times as much space as the water from which it is made. If cold water in sufficient quantities is injected into the exhaust steam from a pump, and the mixing takes place in an air-tight chamber, a vacuum will result, because the exhaust steam is reduced to $\frac{1}{1642}$ its former volume. The degree of vacuum resulting from this process depends on the amount and temperature of the water used. Water and steam contain a certain amount of air, which is non-condensible at ordinary temperatures; therefore, air will gradually accumulate in the condensing chamber in sufficient volume to destroy the vacuum, unless some means, such as a pump, is employed to remove the air as fast as it is released from the steam and water. This air should be reduced to a comparatively low temperature, so as to reduce it to as small a volume as possible before its removal, in order to assist in maintaining a high efficiency in the apparatus.

45. In the arrangement of barometric condensers, the condensing chamber is elevated to such a height that the column formed by the discharge water is sufficient to overcome the atmospheric pressure. This eliminates pump troubles due to handling hot water, but usually makes it necessary to pump the cooling water into the condenser head. In places where the level of the cold water supply is at a sufficient height above the hotwell level, a pump for this purpose is not necessary, the condenser lifting the water by virtue of the vacuum maintained within it. Should the level of the cold-water supply be of sufficient height to enable the condenser to lift its water without the use of a pump, no priming is necessary to start the condenser, as the air pump will create the vacuum necessary to lift the water and start the condenser. With equipment of this kind it is not necessary to place the vacuum pump or the water supply pump close to the condenser. They may be placed within 200 or 300 feet of the condenser chamber.

46. A sectional view of a barometric condenser is shown in Fig. 22. The condensing water enters at *a* and flows into an annular reservoir *b* that has vertical slots in its inner wall through which the water falls in a large number of small streams *c* in the direct path of the exhaust steam, which enters at *d*. The water falling from the annular distributor *b* falls on the tray *e* immediately beneath, and is redistributed by small openings in this tray so as to fall in a fine rain to the bottom of the condenser. The non-condensible gases are drawn from the chamber by a dry-air pump connected at *f*; also the water in falling down entraps some air, which is carried out with the water through the converging nozzle *g* to the atmosphere.

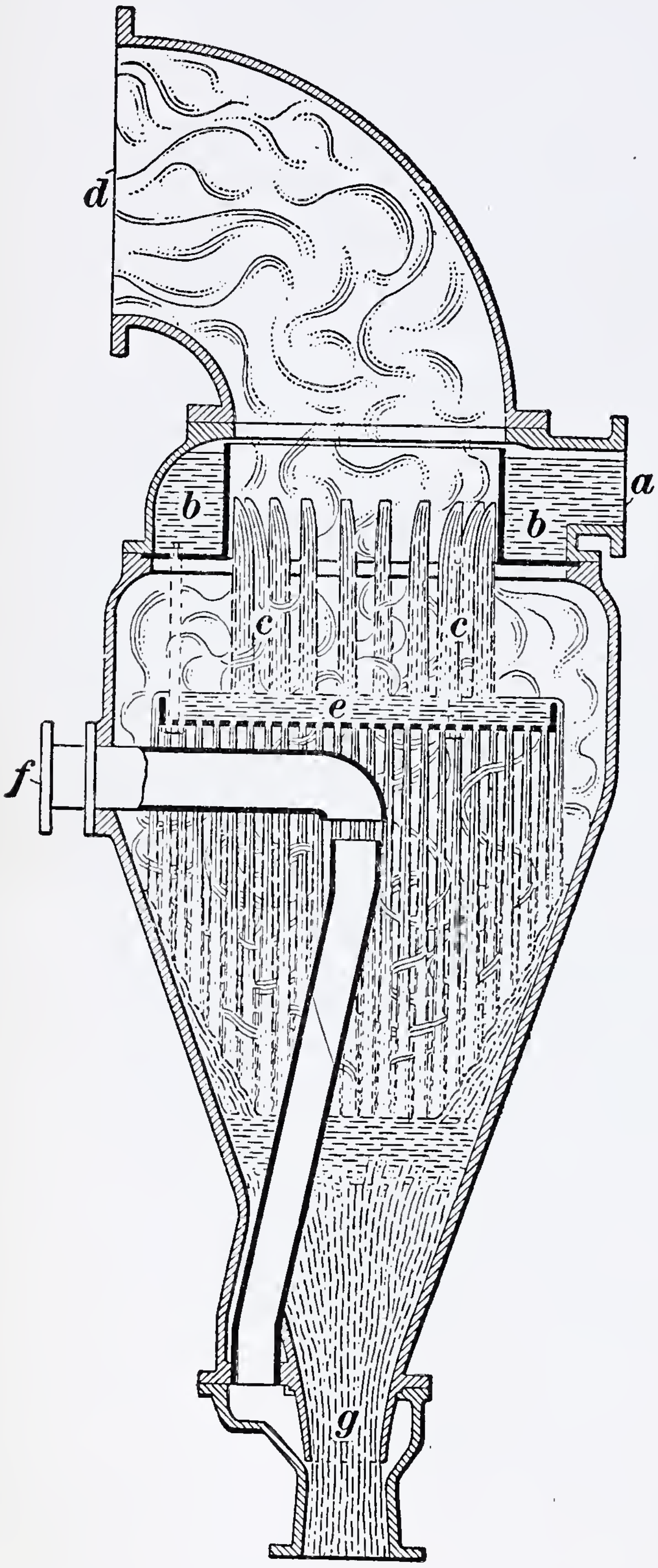


Fig. 22

47. The general arrangement of a barometric condenser is shown in Fig. 23, in which *a* is the source of condensing water and *b* is a pump that lifts the water and delivers it into the condenser chamber *c*. From this chamber the water with the condensed steam and some entrained air falls by gravity through the tail pipe *d* to the hotwell *e*, and thence through the overflow *f* to the atmosphere. The exhaust steam from the pump enters the condensing chamber *c* through the pipe *g*, an atmospheric relief valve *h* being provided to allow the steam to escape to the atmosphere in case the condenser should, for any reason, fail to act. The non-condensible gases resulting from the combining of steam and cold water in the condenser

chamber rise to the top of the chamber and are drawn off through the pipe *i* by the dry-air pump *j*. The distance from

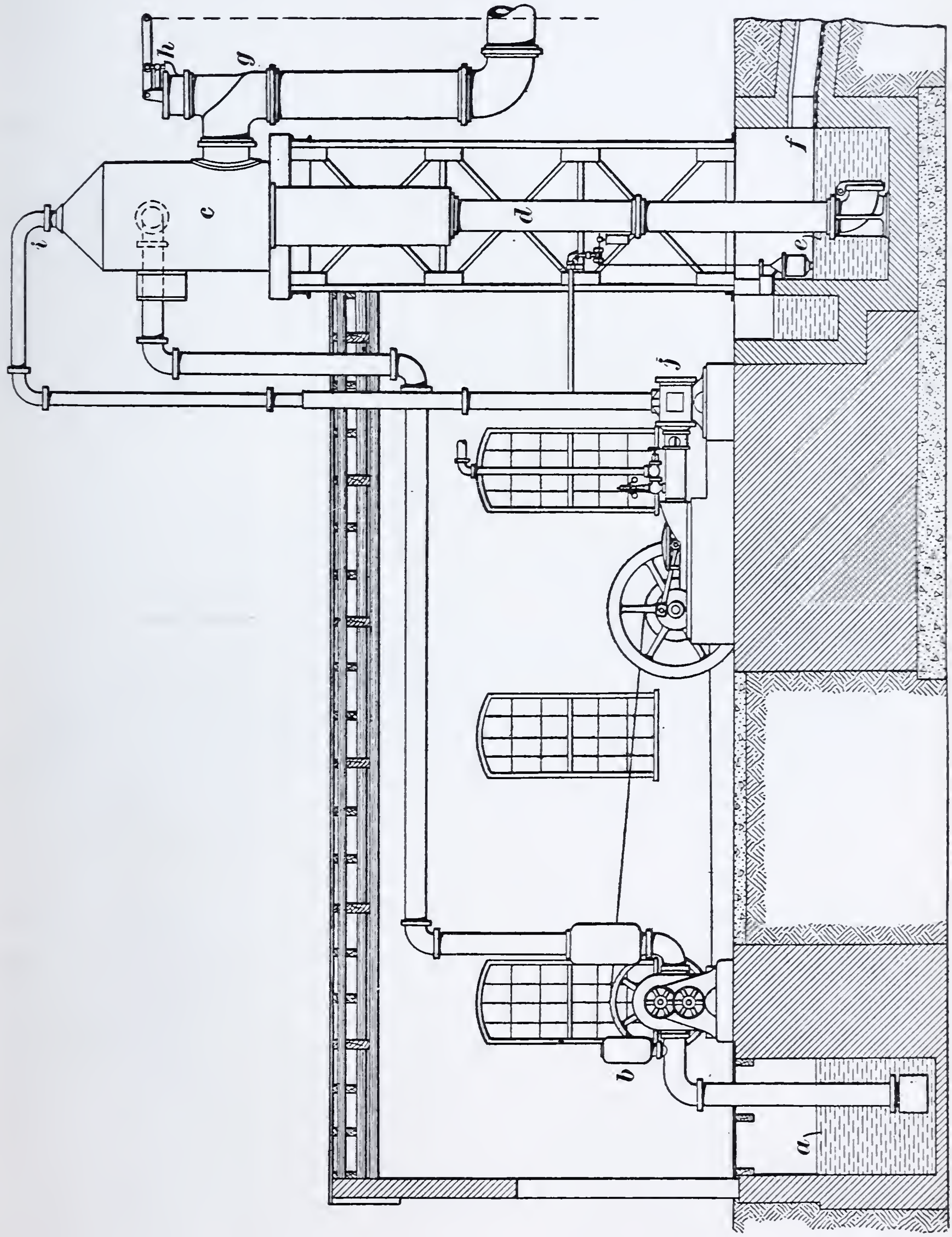


Fig. 23

the surface of the water in the hotwell *e* to the center line of the exhaust steam inlet *g* should never be less than 40 feet.

If the vertical distance from the surface of the water in the cold well *a* to the point of delivery into the condensing chamber does not exceed 18 feet the supply pump can be dispensed with, as the vacuum formed by the dry-air pump will then serve to draw the water into the condenser chamber and thus start the condensing process. This type of condenser requires less water than any other, resulting in a considerable saving of power, especially when the pump *b* can be dispensed with. It has greater stability and accommodates fluctuating loads better than the jet condenser. Also, it is absolutely safe as regards flooding cylinders, and gritty or acid circulating water can be used.

48. Choice of Condensers.—The type of condenser to be used depends almost entirely on local operating conditions, the

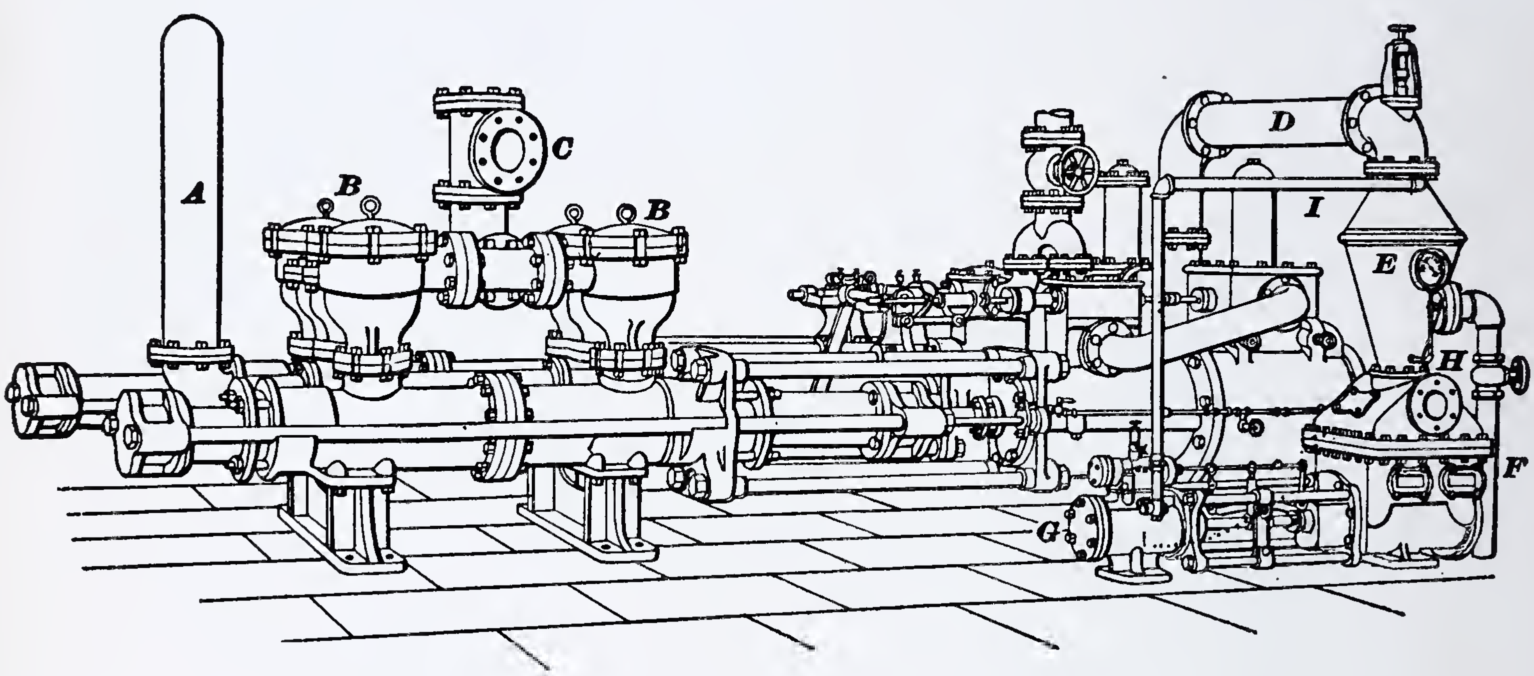


FIG. 24

nature and quantity of boiler feed, and the cooling water available. When cooling water suitable also for boiler-feed purposes can be obtained, some form of jet condenser may be used; if sufficient head room is available, the barometric type of condenser usually proves the more efficient and economical, and can be operated with dirty or acidulous water, if necessary. For high vacuums, and when the available cooling water is unfit for boiler feed, and no suitable and cheap boiler feed can be obtained, the surface condenser finds its most efficient and useful field.

49. Compound Condensing Pump.—A Knowles compound, condensing, outside-packed, duplex mine pump is shown

in Fig. 24; the arrangement of the steam and pump cylinders is similar to that of the pump illustrated in Fig. 15. There are four discharge- and four suction-valve chambers. The suction pipe is placed between the pump cylinders; one end reaches to the sump; the other end has an air chamber *A* bolted to it, which serves to insure a steady delivery to the suction valves. The discharge valves, contained in the chambers *B*, are of the pot-valve type; they work on gunmetal composition seats and are said to be very durable. The delivery pipe is bolted to the flange *C*. The exhaust steam passes from the low-pressure cylinders through the pipe *D* to the condenser *E*, which is of the type shown in Fig. 20, and to which the condensing water is led by the pipe *F*. The mingled condensed exhaust steam and condensing water is discharged by the air pump *G* into the sump through a pipe bolted to the flange *H*. The exhaust steam from the air pump passes through the pipe *I* to the condenser. The pump shown in Fig. 24 is designed to discharge 1,000 gallons per minute against a head of 800 feet.

FLYWHEEL PUMPS

50. Object of Flywheel.—Although direct-acting steam pumps cannot be excelled in simplicity, low first cost, and small expense for repairs, yet they can never be extremely economical in their consumption of steam, even when built compound or triple expansion. By the use of a flywheel, together with the use of suitable valve gears, steam may be cut off at the most economical point in the stroke, and the surplus energy imparted to the steam piston during the first part of the stroke will be stored in the flywheel, to be given up toward the end of the stroke, thus furnishing a nearly uniform driving force for the pump piston or plunger; and this cannot be done in the ordinary direct-acting pump. Flywheel pumps of the direct-acting type are not very extensively used about mines, but there are a number of them in use both at the ore mines of the far West and in the anthracite mines of Pennsylvania.

In a direct-acting pump without a flywheel, the stroke of the steam piston is cushioned by the steam in the exhaust

end and if there is any water in the cylinder the pump will be stopped. With a flywheel, however, the effect of the cushioning is overcome by the energy stored up in the flywheel and the engine is subjected to shock. A pumping engine usually runs only intermittently and for this reason an engine without a flywheel is generally preferred.

51. A small single Cameron pump having a crank and flywheel is shown in Fig. 25. This pump has a crank *a* attached

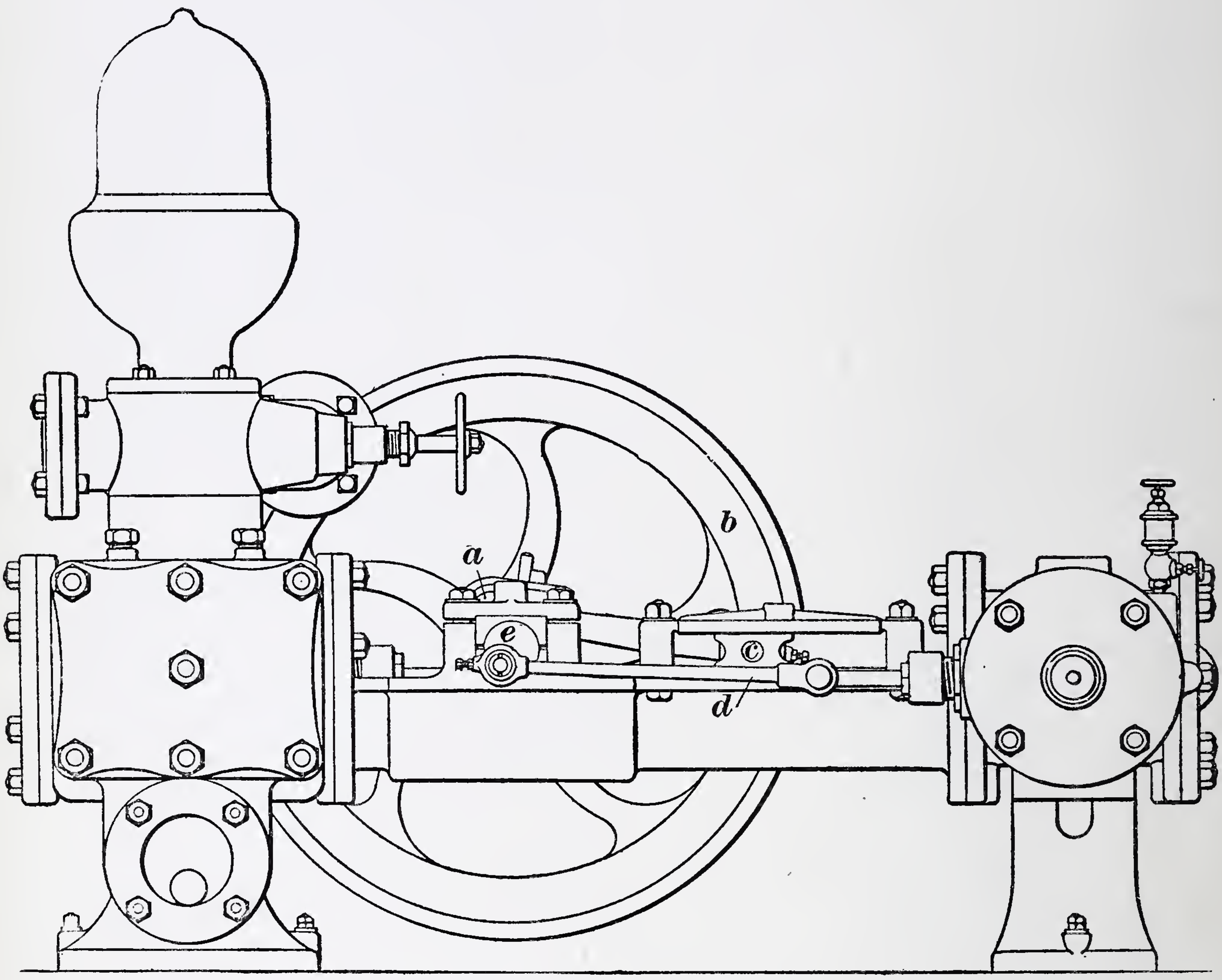


FIG. 25

to a flywheel *b* and to a crosshead *c* by means of a connecting-rod. The water piston and the steam piston rods act in a direct line and the valve rod *d* is moved by the crank *e* that is operated by the flywheel *b*.

52. In Fig. 26 (*a*) is shown a side view of the high-pressure side of a duplex pump driven by a cross-compound, condensing, Corliss engine provided with a flywheel and double plunger. Fig. 26 (*b*) is an end view of the water end of both pumps, looking toward the engine, and Fig. 26 (*c*) is an end

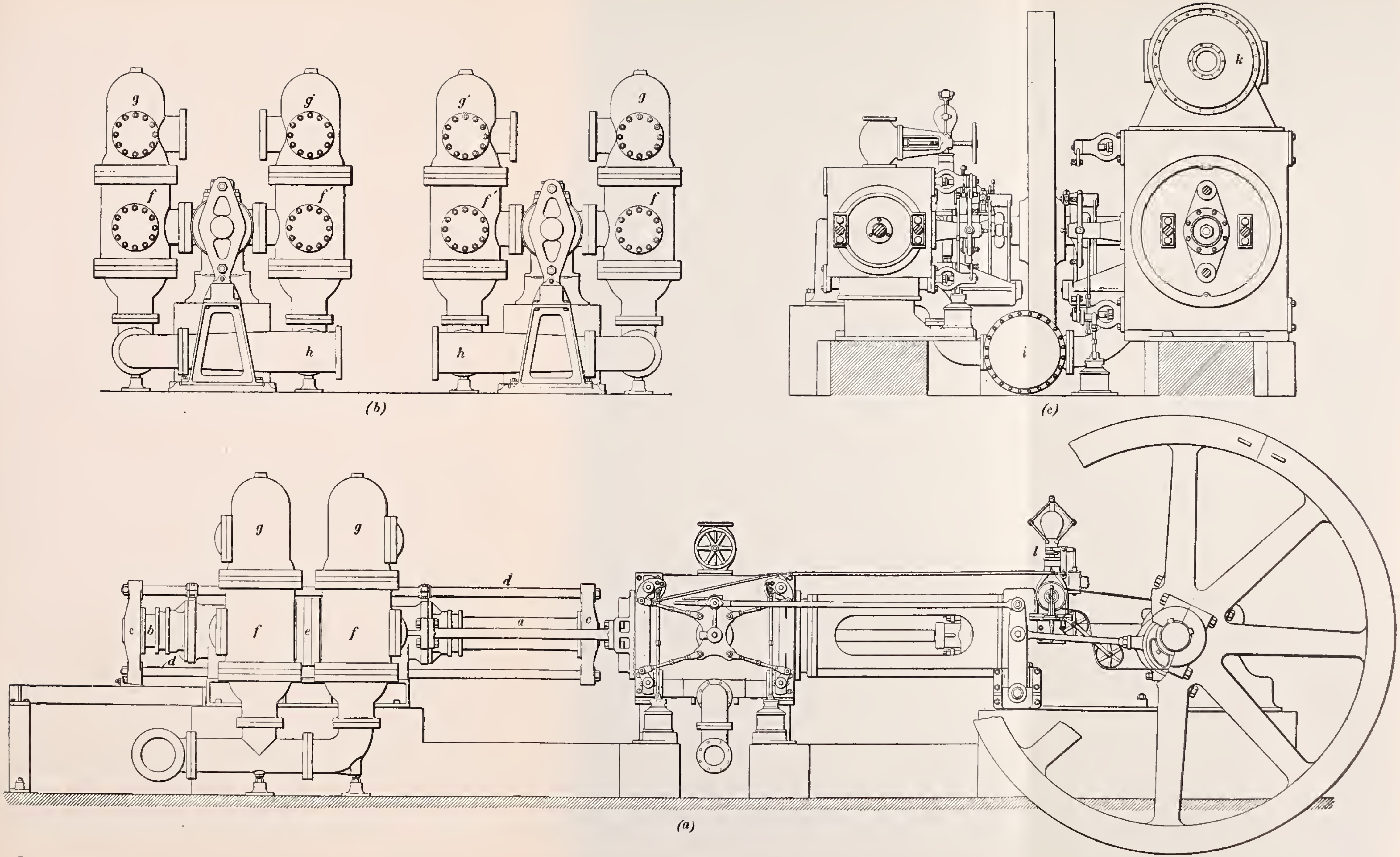


FIG. 26

view of the engine, looking toward the flywheel, the observer being supposed to stand between the pumps and the engine. The plungers *a* and *b* are connected by yokes *c* and rods *d*, and are driven directly by the piston rods of the high-pressure and low-pressure cylinders, which for this purpose are prolonged beyond the pistons and pass through the back cylinder heads.

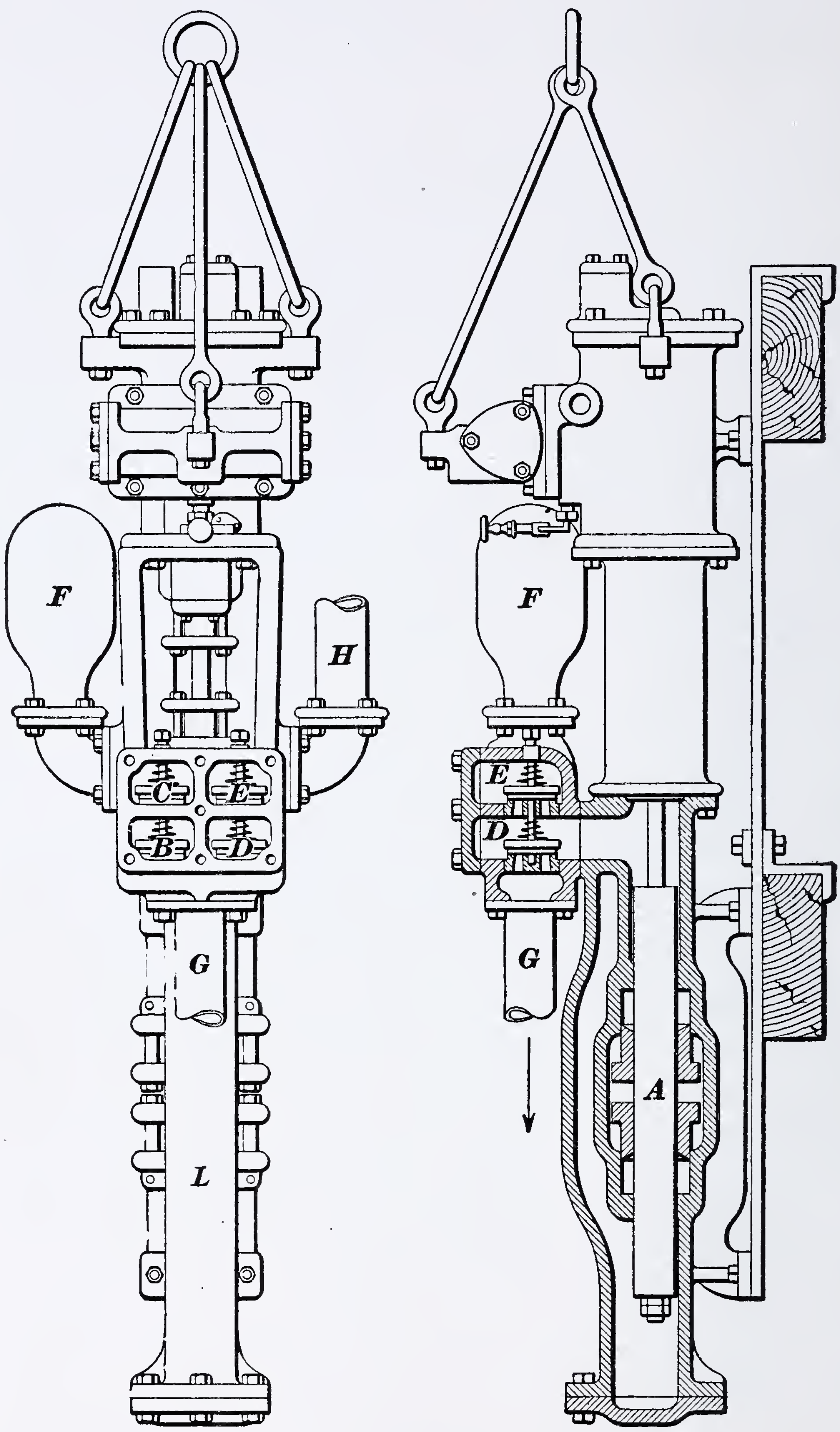
53. The pump cylinders Fig. 26, are divided by a diaphragm *e* in the center, and each pump cylinder has two suction valve chambers *f, f′* and two delivery valve chambers *g, g′*. These valve chambers are placed on both sides of the pump cylinders. The four suction-valve chambers of each pump connect to the suction branch *h*, and the two branches in turn are connected to the suction main by a **Y** fitting not shown in the illustration. The four delivery chambers of each pump are connected together by branch pipes, and these branch pipes, in turn, discharge into a common main delivery pipe.

54. A reheating receiver *i*, Fig. 26, is placed between the high-pressure and low-pressure cylinders. The low-pressure steam-inlet valves are placed beneath the low-pressure cylinder; the low-pressure exhaust valves are on top and exhaust directly into the chamber *k*, which is placed on top of the low-pressure cylinder. The high-pressure valves are arranged in the usual way. The engine is provided with a variable speed governor *l*, by means of which the speed of the engine may be varied to suit the requirements of the service.

The particular pump illustrated has cylinders 32 and 60 inches in diameter and a 48-inch stroke, the plungers being $13\frac{3}{4}$ inches in diameter. Each pump is designed to lift 3,500 gallons of water per minute against a head of 550 feet. The water to be pumped is highly charged with sulphuric acid and, to guard against corrosion, the pump cylinders, valve chambers, and all pipes are lined with lead and the plungers and valves made of acid-resisting metal.

SINKING PUMPS

55. Steam- or Air-Driven Sinking Pumps.—For sinking purposes, single, duplex, or triplex, direct-acting, plunger



(*a*) Fig. 27 (*b*)

pumps driven by steam or compressed air have, heretofore, been generally used. A sinking pump must be as light as possible, consistent with strength, so that it can be readily

raised or lowered by ropes as the shaft is gradually deepened. The working parts should be housed so that they will not be damaged by flying or falling rock. Since they must generally pump gritty water, the valve openings should be large and the valves so placed that they may be readily inspected and replaced when necessary. All parts of a sinking pump should be made as simple as possible, so that there will be the least possible likelihood of their getting out of order and so that they can be speedily repaired if necessary. Sinking pumps are generally double acting.

56. In Fig. 27 (*a*) and (*b*) is shown a double-acting, plunger, sinking pump. When this pump is working and the plunger is moving downwards, the water is forced out of chamber *L* through the valve *C*, into the discharge pipe *H*. The water enters the space left by the downward-moving plunger from the suction pipe *G* and in so doing raises the valve *D*. When the upward stroke takes place, the valves *C* and *D* close, and, the valve *E* being opened, water is forced into the discharge pipe *H*. The movement of the plunger opens the valve *B* and water is drawn into the chamber *L* through the suction pipe. In the figure, *F* is the air chamber. The section shown in Fig. 27 (*b*) is taken through the center line to show the plunger, stuffingboxes, etc., and the valve section is taken on the center line of the valves *E* and *D* in order to show better the details of the principal parts. Spiral valve springs are shown above the valves, their object being to seat the valves.

PUMP DETAILS

PISTONS AND PLUNGERS

57. Methods of Packing.—An *inside-packed pump* is a pump in which the packing is inside the water cylinder, or working barrel. All piston pumps and some plunger pumps are inside packed, as shown in Figs. 6 and 13, respectively. As a rule, inside-packed pumps should be avoided for mine

use, because acid water and water containing gritty material wear out the packing rapidly and cause leaks that are not easily discovered. Further, to inspect the condition of the packing requires that the pump be shut down and the cylinder head removed; a process that takes more time than can possibly be spared in mining where water is continuously flowing into the workings. An *outside-packed pump* is a pump in which the packing is outside the water cylinder or barrel. An *outside end-packed pump* is a pump in which the stuffingboxes are at the outside ends of the water cylinder, as shown in Figs. 8 and 15. An *outside centrally-packed pump* is a pump in which the packing is outside the water cylinder, but between the ends of the two water cylinders as in Fig. 9. Outside-packed plunger pumps are best adapted to mining use under heavy pressures.

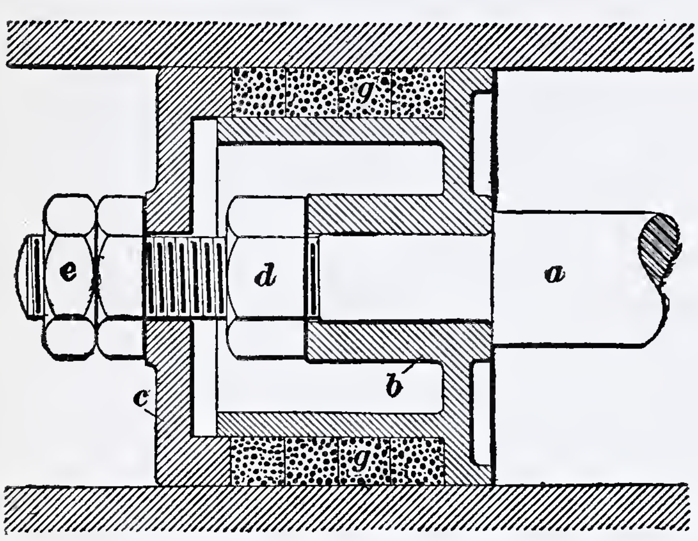


Fig. 28

58. Water Pistons.—A water piston, in section, fastened to a piston rod *a* is shown in Fig. 28. The box *b* with the gland *c* forms a piston and gland for the square fibrous or rubber packing *g*. The box *b* is held in place by a shoulder on the piston rod *a* and by the nut *d*; and the gland *c* is held by the nuts *e*. By screwing down the nuts *e*, the gland is forced against the packing, causing it to spread and fill the space between the piston and the cylinder, thus making a water-tight joint with the inside of the cylinder. When the packing wears, it may again be tightened by screwing down the nuts *e*. There are numerous types of water pistons, but this description covers the principle of practically all of them.

59. Water Plungers.—Pump plungers are made both solid and hollow; in either case they are turned in a lathe and polished so as to work smoothly in the stuffingbox and not cut the packing. Usually, large-sized plungers are made hollow in order to reduce their weight, thus, because of their lightness, reducing the wear in the stuffingboxes. In Fig. 29 are shown

two plungers and the method of fastening them so that they will offer a solid end to the water to be displaced. In Fig. 30 is shown a plunger *a* attached to the plunger rod *b*. The packing ring *c* for the plunger is bolted to the lugs *d* and the hemp packing *e* is kept in place by the gland *f* and the bolts *g*. Any wear on the packing is taken up by screwing up the gland *f*. In order to tighten up or repair a packing box placed on the inside of the cylinder, as shown in Fig. 30, it is necessary to remove one or both ends of the water barrel; as such repairs cannot be quickly done, this method of packing, as previously stated, is not well adapted for mine pumps. An outside end-packed plunger is shown in Fig. 31. The plunger cap *a* is made of an acid-resisting metal, while the plunger *b* is made of cast iron, it having been found in mining work that the plunger cap or point is the only part that is attacked by acid water. This is because the end of the plunger is in the water all the time while the body of the plunger is in the water only part of the time and, further, the play of the latter through the stuffingbox and its usual coating of grease serves to protect its surface from corrosion. The grease ring *c* fits into the stuffingbox and is placed between the rings of fibrous packing. It is recessed both inside and outside and has several holes by which the outside recesses connect with the inside recesses. The outside recess is in connection with the grease cup *d*, which is provided with a cock. When it is desired to grease the plunger, the cock is opened and the

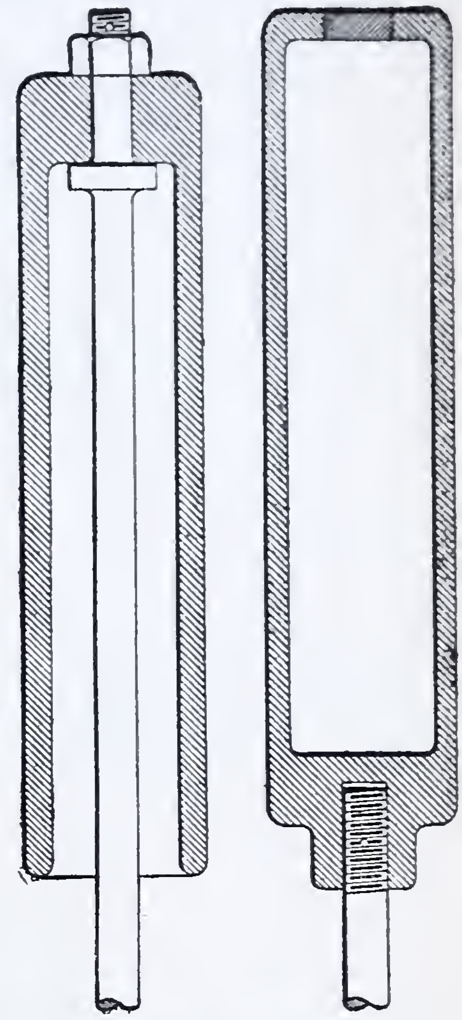


FIG. 29

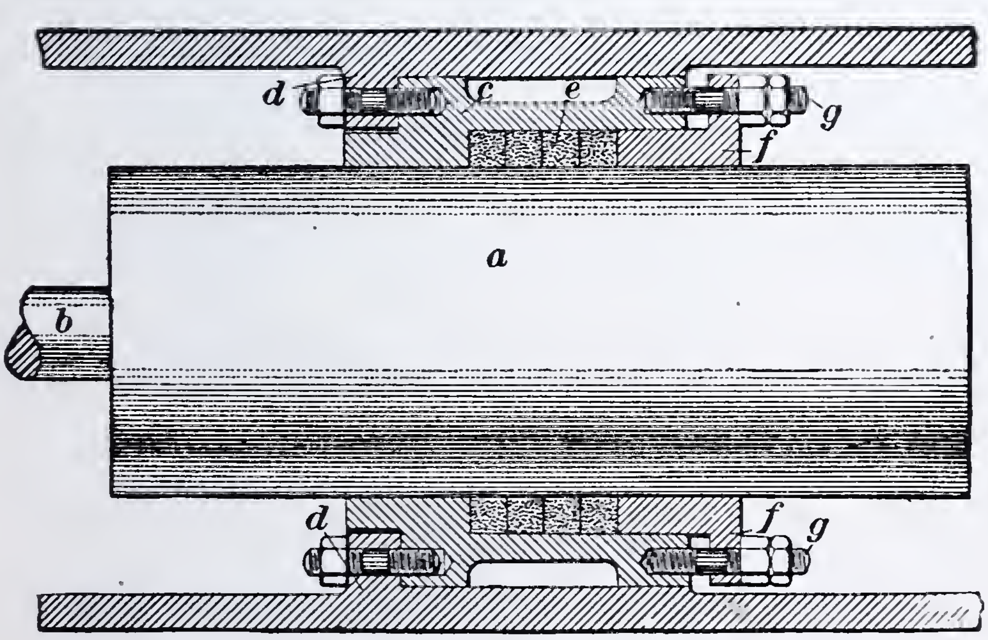


FIG. 30

grease is forced into the space around the grease ring by the screw *e* on top of the grease cup. This is done once or twice during the day, and the cock is then closed so as to relieve the

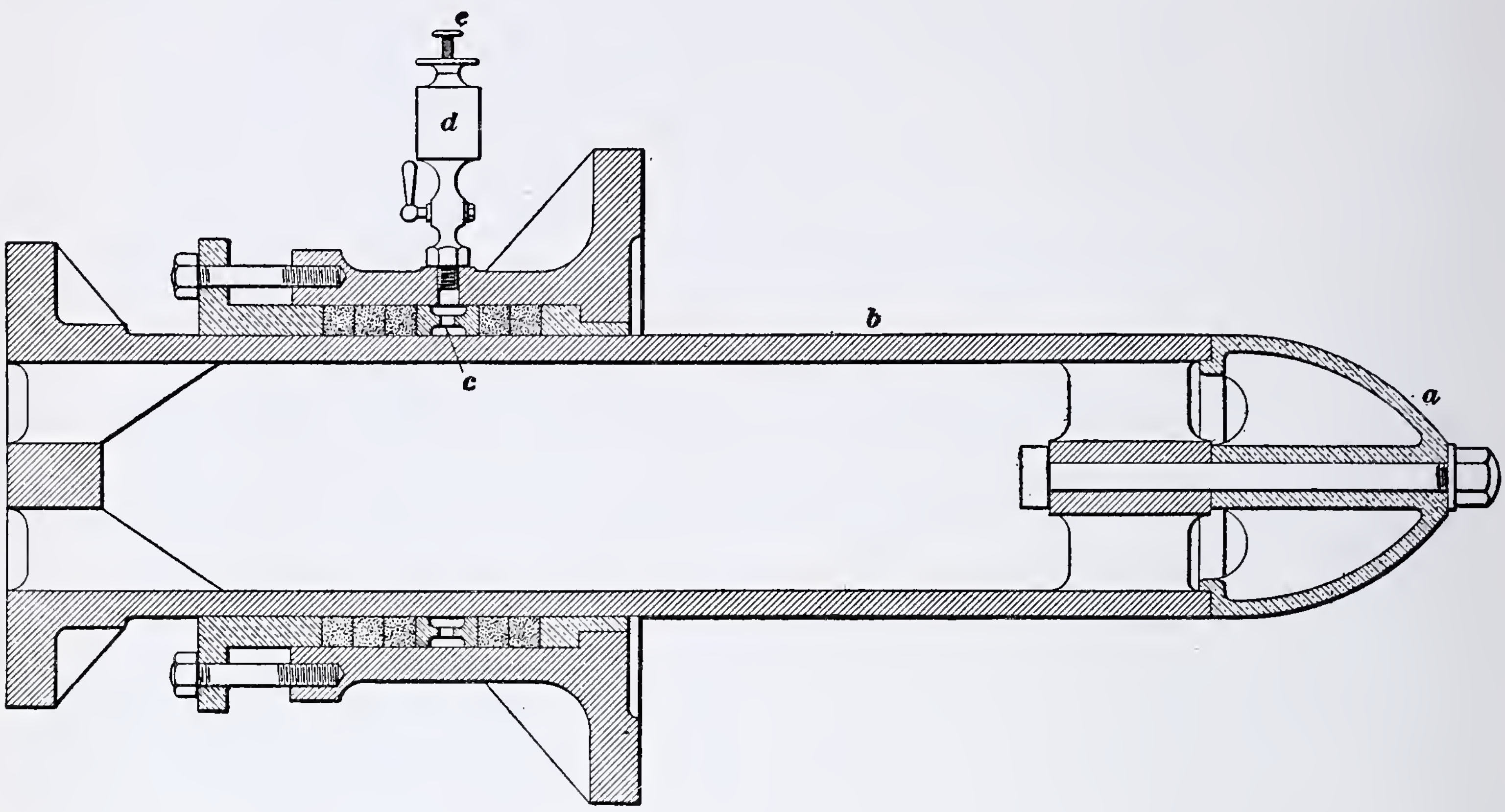


Fig. 31

grease cup of the water pressure and to prevent consequent leakage. The stuffingbox is bolted directly to the pump cham-

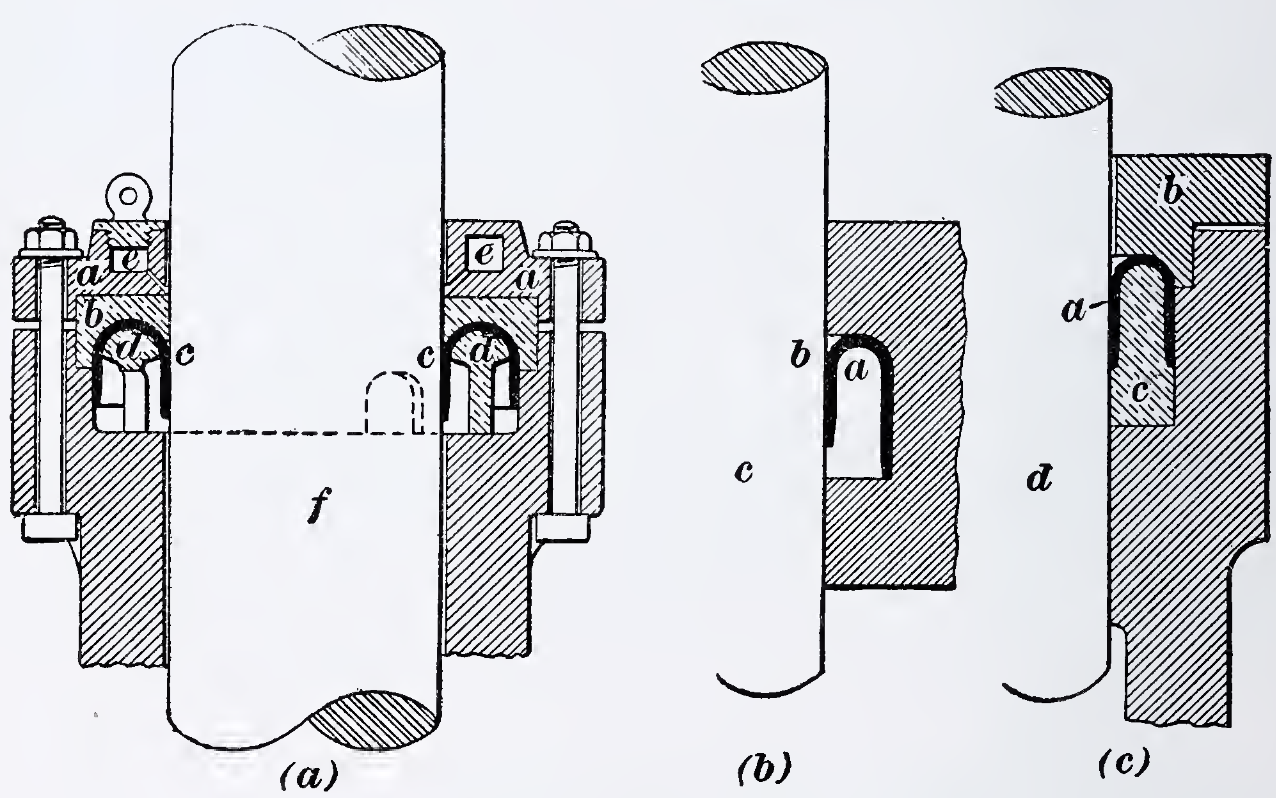


Fig. 32

ber, which may be any type, but for high-pressure mine work it is generally circular.

60. Three methods of using leather packing for very heavy pressures are shown in Fig. 32. View (*a*) shows a gland *a* lined with a brass ring *b*, which holds the cup leather *c*

down on a brass supporting ring *d*. A chamber *e* in the gland holds the grease for lubricating the plunger *f*.

View (*b*) shows the cup leather *a* held in a recess cast in one end of the pump cylinder. While this method of packing the plunger is inexpensive, there is difficulty in casting a recess that will fit the leather. The plunger must be removed when a new leather is inserted, which makes its packing difficult. Experience shows that the leather wears at the bend *b*, where it presses with greatest force against the plunger *c*.

View (*c*) shows a cup leather held in position by a gland *b*, over a brass ring *c* so fitted as to prevent severe pressure against the plunger *d*. If the gland is accurately turned to bear against the curved portion of the leather, it will form a better support and increase the life of the packing.

PUMP VALVES

61. General Considerations.—The successful operation of reciprocating pumps depends largely on their valves; therefore, certain important factors must be known before the valves are designed, such as the impurities that are in the water, which will determine the metal to be used in the valve seats; whether the water is to be hot or cold, which will determine the material to be used in the valve; also the speed with which the pump will run and the pressure to be overcome, since high-speed pumps, or pumps having a piston travel greater than 100 feet per minute, will require larger valve openings than slower-speed pumps, and in no case should the water flow through the valve passages at a greater velocity than 240 feet per minute. Again, pumps that are to be subjected to heavy pressures must have strong material in their valves. All valves are subject to wear and shock, which makes it imperative that the pump be so constructed that, if necessary, the valves can be readily replaced.

62. Location of Valves.—In a lift pump, the valves are placed both in the piston and in the suction and discharge pipes. In a force pump of the vertical type, the valves are usually

placed in the suction and discharge pipes; but in a force pump of the horizontal type, the valves are placed in separate chambers. The portion of the pump that holds the valve seats is known as the *suction-valve deck,* or *discharge-valve deck,* in accordance with the kind of valves contained.

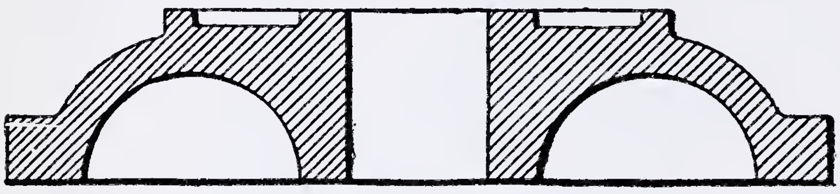


FIG. 33

63. Materials of Construction.—Pump valves are made of various materials, according to the temperature, purity, and pressure of the water. Metal or some other material not affected by heat is used when hot water is to be pumped. Metal valves similar in cross-section to that shown in Fig. 33 are sometimes adopted for very heavy pressures. Bronze or rubber is used when acid water is to be pumped. Hard rubber is very serviceable for moderately heavy pressures, but it will chip and break with very heavy pressures, and in such cases metal is preferable. Soft-rubber or leather valves do not wear well, as the pressure forces them into the water spaces in the valve seat, and this, continued with each stroke, cuts and renders them unfit for service. Wooden valves have been used in very acid mine water, but they wear rapidly.

64. Disk Valves.—A disk valve that is much used in horizontal pumps for ordinary pressures is shown in Fig. 34. In the figure, the valve *v,* of vulcanized rubber, rests on a metallic valve seat *s* provided with threads *t* to screw into the valve decks. The cage *g* is for the purpose of guiding the spindle *o* and regulating the lift of the valve. To prevent the valve from coming in contact with the cage *g,* a cap *p* is fastened to the spindle. There are no springs in this design, which is somewhat in its favor, since springs break, become weak, and wear out. This valve will not seat as quickly as a spring-actuated valve and will be more subject to shocks, since, where a spring is not used, an interval will exist before the action of the water tends to seat the valve, and the valve will seat with a shock.

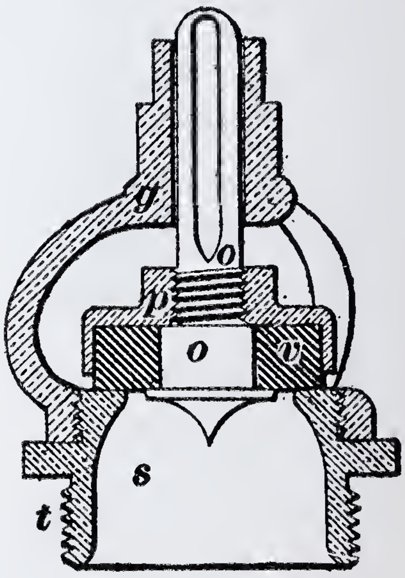


FIG. 34

65. In Fig. 35 is shown a common form of rubber disk valve *a* on a spindle *b* held in place on the valve seat *c*, by the spiral spring *d*, cap *e*, and nut *f*. Above the valve, there is a metal disk *g* that prevents the spring from wearing the rubber. In case the spring is made heavier and helical, the cap *e* is made with wings to receive the spring. Disk valves, Figs. 34 and 35, vary in size from 2 to 6 inches in diameter, the most common size for ordinary conditions being about 3 inches.

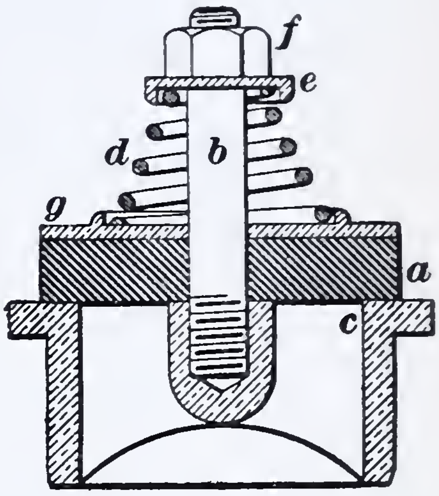


FIG. 35

66. The section of a large disk valve and seat sometimes adopted for mine pumps is shown in Fig. 36. The valve seat *a* is held in place to the proper deck by the flange *b*. The top of the valve seat, shown in plan, has a number of passages through which the water flows when the valve *c* is raised. Above the valve is a metal cap *d* that, with the valve, slides on the valve stem *e*. Between the head of the valve stem and the metal plate, a spiral spring *f* is placed, and then the valve stem is screwed into the valve seat.

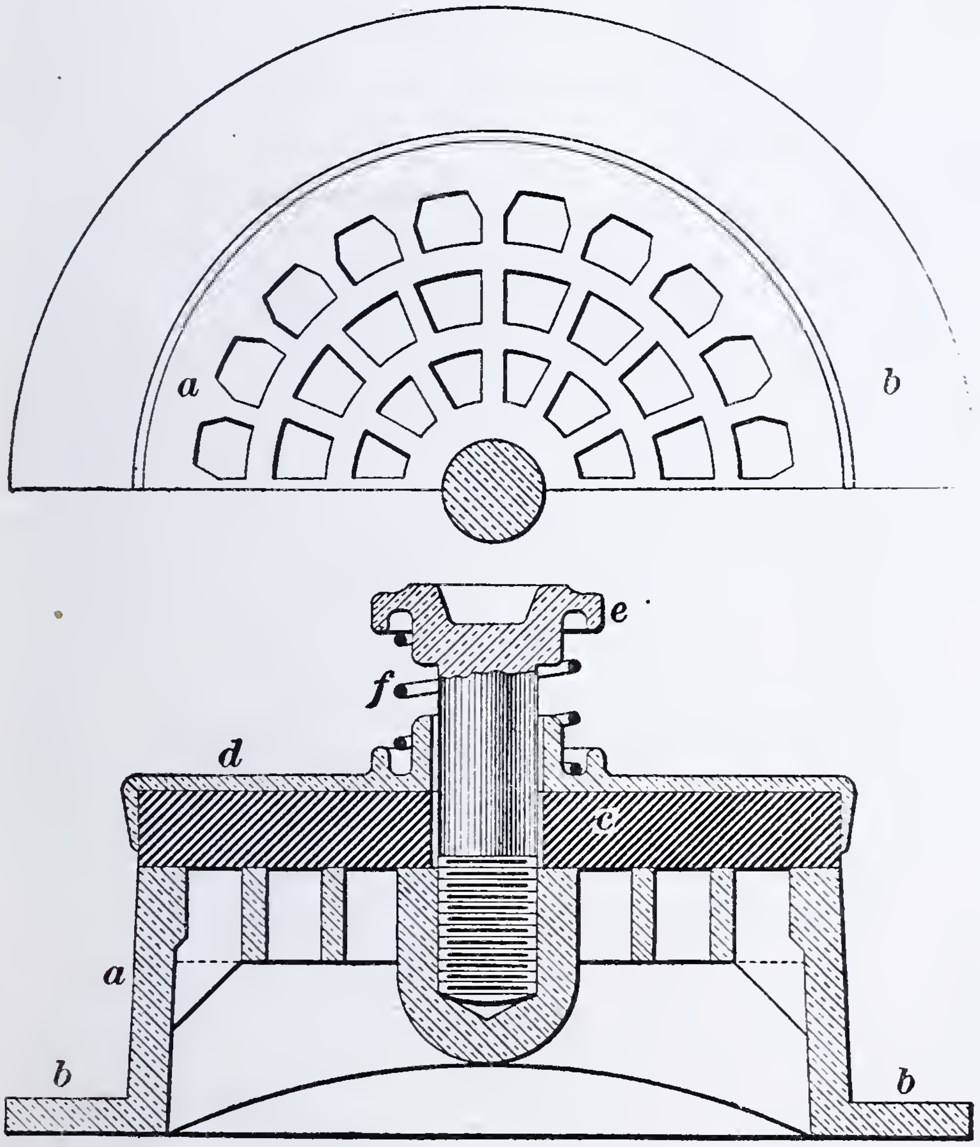


FIG. 36

67. A section of a lift-pump piston and disk valve is shown in Fig. 37. The valve stem *s* is threaded for a nut to hold the valve and seat in place. The hard-rubber valve *a* is partly covered by a metal disk *b*; above this, a spiral spring *c* regulates the lift and assists the valve in seating quickly. The cap *d* that

holds the spring to its seat is held in place by a jam nut *f*. Another lift-pump piston, shown in Fig. 38, is composed of a disk valve *a*, valve seat *b*, and yoke of metal *c*, held in place by nuts *d* and *e* on the piston rod. This valve is intended for higher pressures than the valve illustrated in Fig. 37. The annular space *f* is filled with packing to form a water-tight joint.

68. Clack Valve.—In Fig. 39 is shown a piston with a circular clack valve *a* composed of rubber and leather riveted

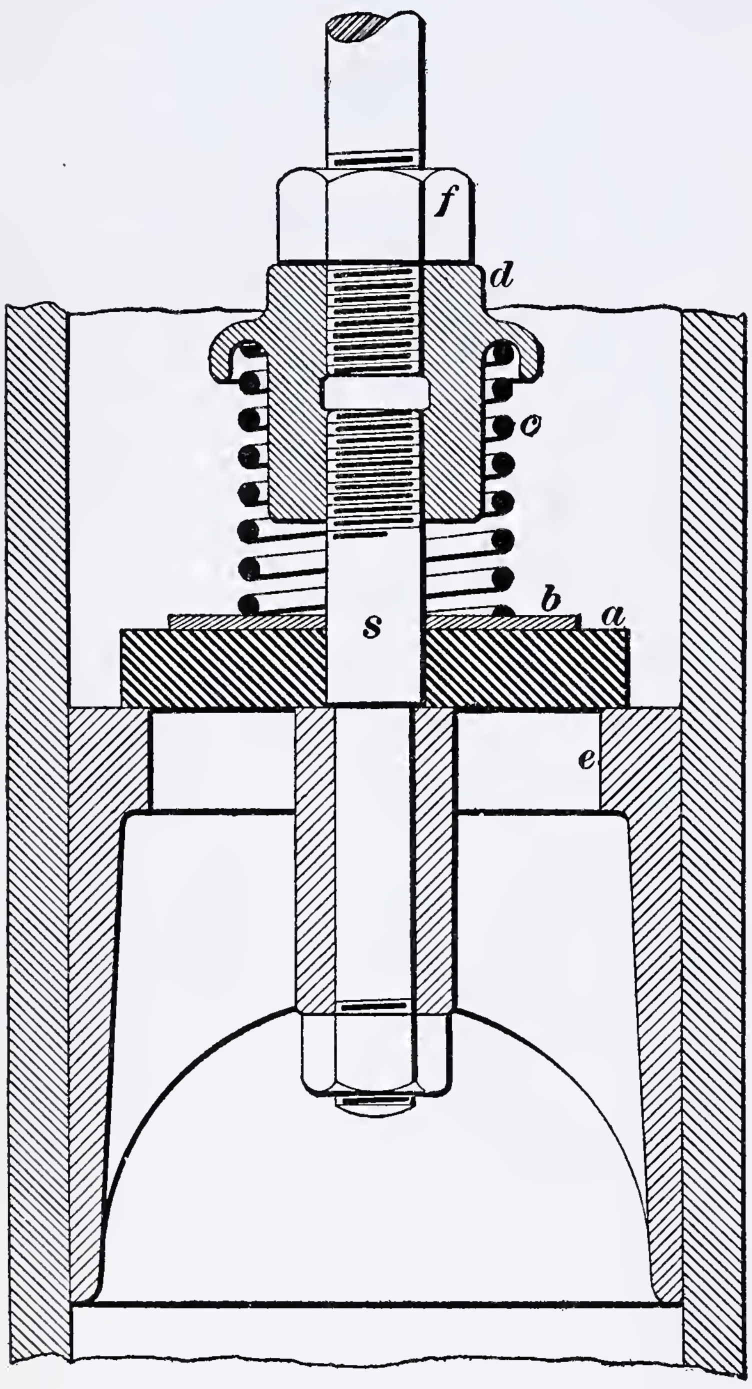


Fig. 37

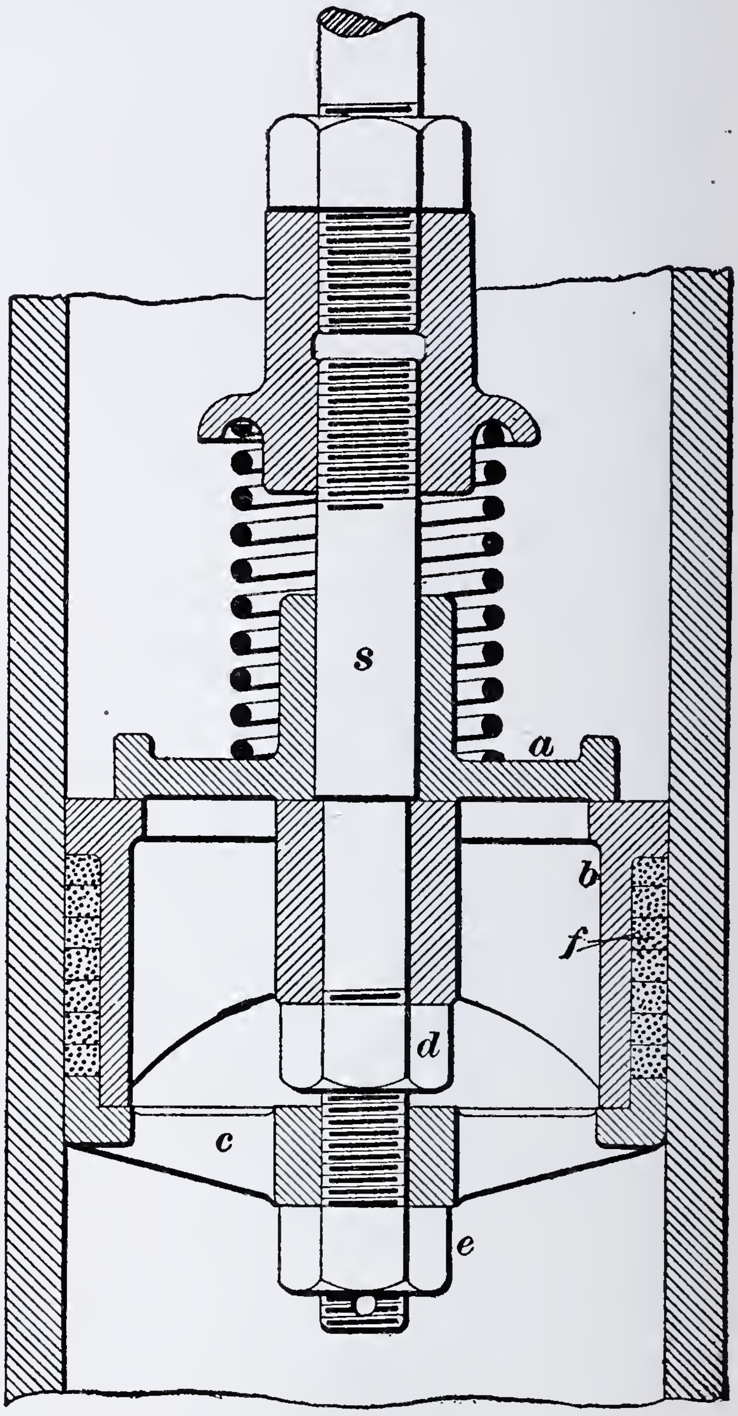


Fig. 38

together. As the piston descends, the valve rises and meets a circular, cup-shaped valve guard *b* held in place by a collar *c*; the piston rod *d* passes through the valve *a*, guard *b*, valve seat *e*, and yoke *f*. The valve seat *e* has a number of passages *g* for the water to pass through, and is reenforced by the wings *h*. The yoke *f* is for the purpose of keeping the valve seat in place, and is constructed so that, in connection

with the valve seat, it forms a stuffingbox *i* into which packing soaked in tallow is placed for lubrication and to make a water-tight joint.

Another style of clack valve in a working barrel above the suction pipe is shown in Fig. 40. The valve *a* is flexible and strikes against the guard *c*, when it lifts, and falls on the valve seat *b*. The objection to this valve is that it rises and falls in the same place, stroke after stroke, and consequently wears more than would be the case if the valve turned slightly at each stroke. Sometimes the water passages in the gridiron *b* are slanted so that the valve will be turned slightly, as the water then strikes at an angle; such an arrangement, however, causes increased friction. Another method is to slant the edges of the valve, or give them inclined milled edges, then each time the valve lifts it will rotate a little on the valve stem and seat in a different place when it falls.

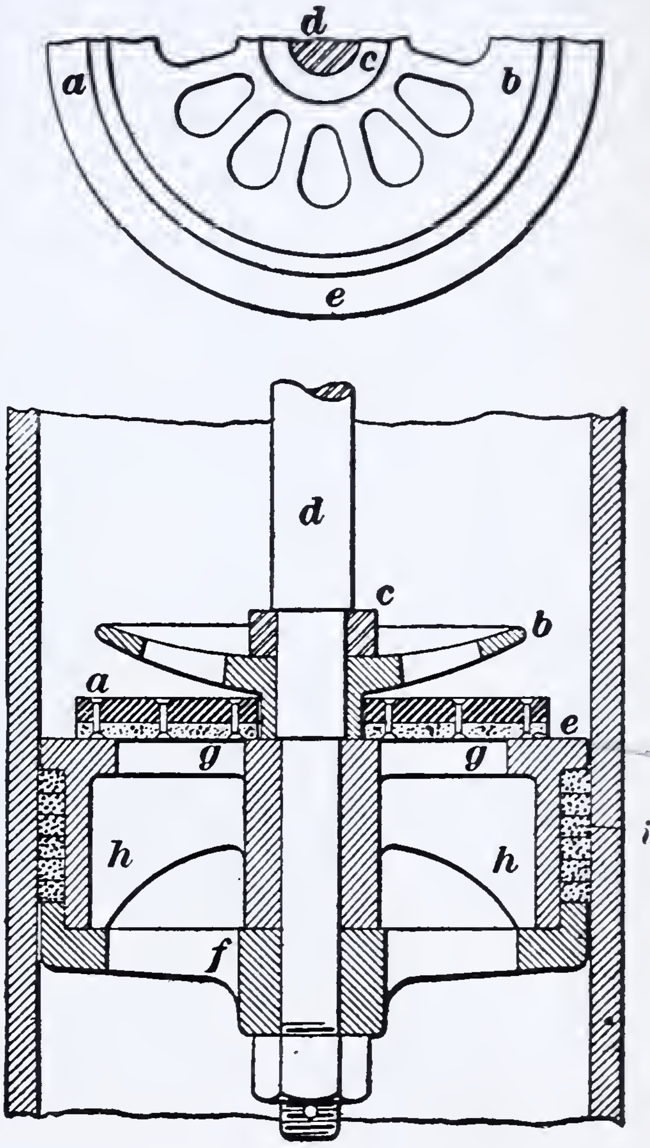


Fig. 39

69. Cornish Valve.—The valve shown in Fig. 41 is known as a *double-seat valve*, and is much used in pumps that work against high pressures. The valve *a* is a sort of casing that has a seat at *c* and *d*. The valve seat is of a peculiar shape and is held to the valve deck by a yoke, or a cage, not shown, that covers the valve and is fastened to the valve deck. The valve slides up and down on the stem *b* and is prevented from lifting too high by the nut *e*. When the valve is lifted, water passes upwards through the openings at *c* and *d* as shown by the arrows. This is one

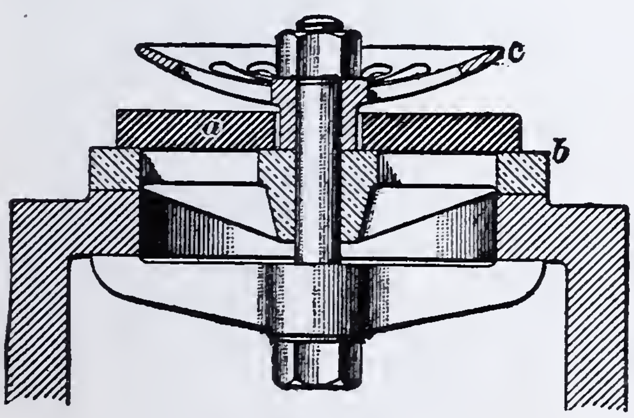


Fig. 40

of the most serviceable valves and has worked satisfactorily under pressures of 350 pounds per square inch.

70. Pot Valves.—The valves shown in Figs. 42 and 43 are used for mine pumps working against heavy pressures and

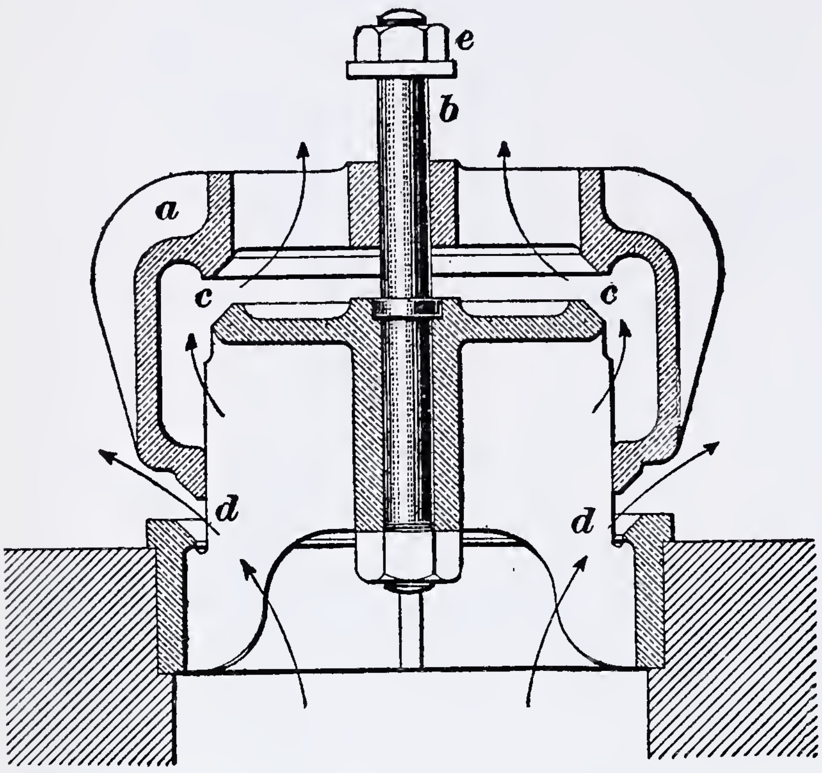


FIG. 41

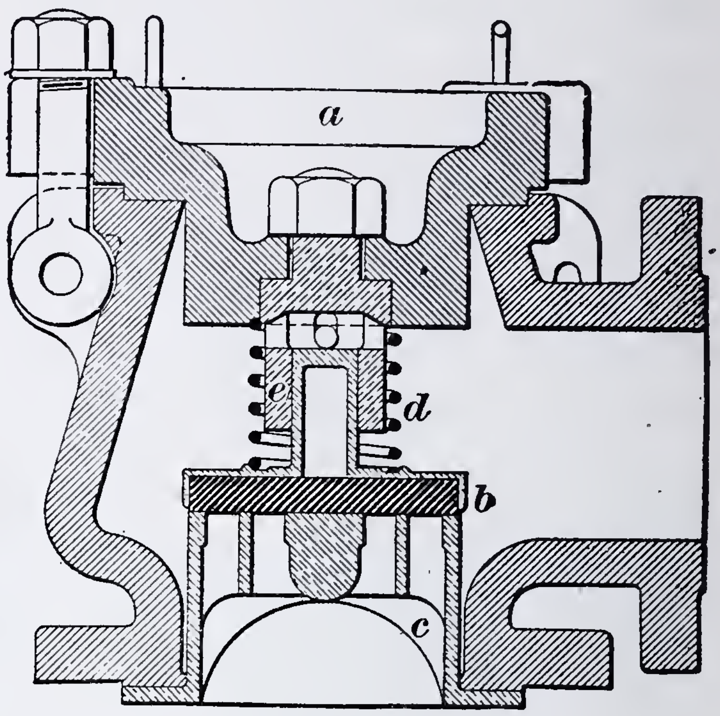


FIG. 42

are known as *pot valves*. Fig. 42 shows a section of a valve made separate from the pump chambers so that it can be replaced when broken or worn. The cover *a* is secured by hinged bolts with nuts, in order that the valve *b* and seat *c* can be reached quickly. The valve guide stem *e* is securely bolted to the cover *a* and the valve seat *c*, thus permitting play to the valve and the valve spring. The parts that come in contact with the water are made of composition metal if the water is corrosive.

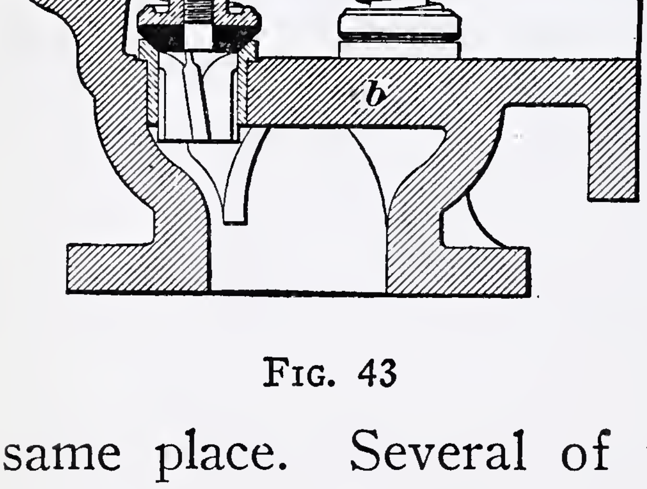


FIG. 43

Another pot valve is shown in section in Fig. 43. This valve is intended for high pressures, and hence is made small, with a beveled seat and winged bottom piece so that it will not seat twice in the same place. Several of these valves are placed in one pot, whose cover *a* is secured to the pump chamber by hinged bolts having nuts. One valve is shown in section, another in elevation, seated on the pot deck *b* and held in position by cover *a*.

AIR CHAMBERS

71. Object of Air Chambers.—Even in double-acting pumps there is an interruption in the flow of water at the end of the stroke, when the piston changes its direction of motion. This has the effect of bringing the column of water in the suction and discharge pipes to rest at the end of each stroke, and this column of water must be set in motion again as the next stroke is made. If the pipes are long, the force required to stop and start the water will be very great, and there will be a severe shock at the end of every stroke that will absorb power and subject the pump and pipes to great stresses.

This difficulty is removed and the flow through the pipes is made more continuous and steady by the use of air chambers. An air chamber is a vessel containing air and is attached either to the pump just outside of the discharge valves, as in Fig. 7, or to the discharge pipe near the pump, as in Fig. 6. While small duplex pumps are often run without an air chamber, it is better in general to fit one to all pumps, since its effect will always be beneficial.

72. Discharge-Pipe Air Chamber.—In Fig. 44, an air chamber *b* is shown attached to the discharge pipe of a single-acting plunger pump *d* for boiler feeding. The water drawn into the pump through the pipe *f* and the valve *g* is forced by the plunger *c* through the valve *h* into the discharge pipe *a*, compressing the air in the chamber *b*. When the plunger reaches the end of its stroke and no more water is being forced into the discharge pipe, the compressed air in *b* forces the water from the air chamber into the discharge pipe. The compressed air in the chamber thus acts as a spring that absorbs the extra force during the forward stroke of the plunger and gives it out during the return stroke, thus relieving the pump and pipe of shocks and keeping the water in motion. Under great pressure, the air in the air chamber is absorbed by the water and gradually carried off. This will render the air chamber useless unless some means is provided for renewing the air supply.

The proper size for an air chamber depends on the type of pump, the speed at which it works, the length of the discharge pipe, and the pressure head against which the pump works. For ordinary double-acting pumps working against moderate pressures and at ordinary speeds, the cubical contents of the air chamber should be not less than 3 times the piston displacement. For pressures of 100 pounds per square inch and

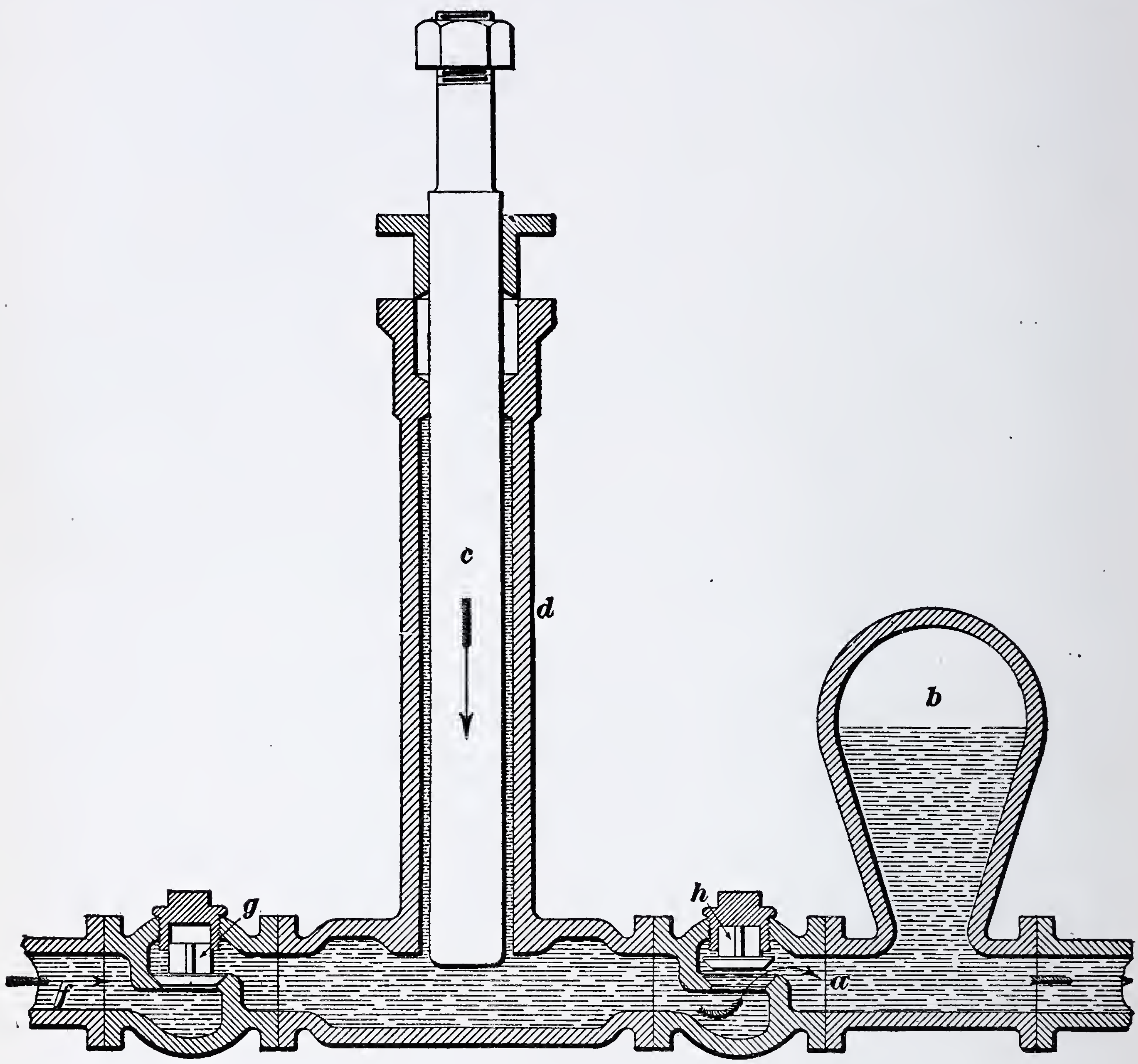


Fig. 44

upwards or for high piston speeds (as in the case of fire pumps) the capacity of the air chamber should be at least 6 times the volume of the piston displacement for a single stroke.

73. Alleviator.—When a pump works under a pressure greater than that due to a 150-foot lift, an air chamber is not of much service, owing to the fact that the air is forced either through the pores of the iron or out at the joints, or is absorbed

and carried off by the water. To obviate this defect, an *alleviator* shown in Fig. 45 is used. It consists of a plunger *a* working through a water-packed stuffingbox. On top of the plunger are arranged alternately rubber buffers *b* and steel plates *e* that are held in place by the tie-rods *c* and the yoke *d*. When the pressure in the column pipe exceeds the working pressure, the plunger *a* is forced out through the stuffingbox and relieves the pump of the shocks that would otherwise occur. Alleviators may be placed anywhere on the delivery pipe, but are preferably placed in such a position that the direction of the moving water is in line with the plunger *a*.

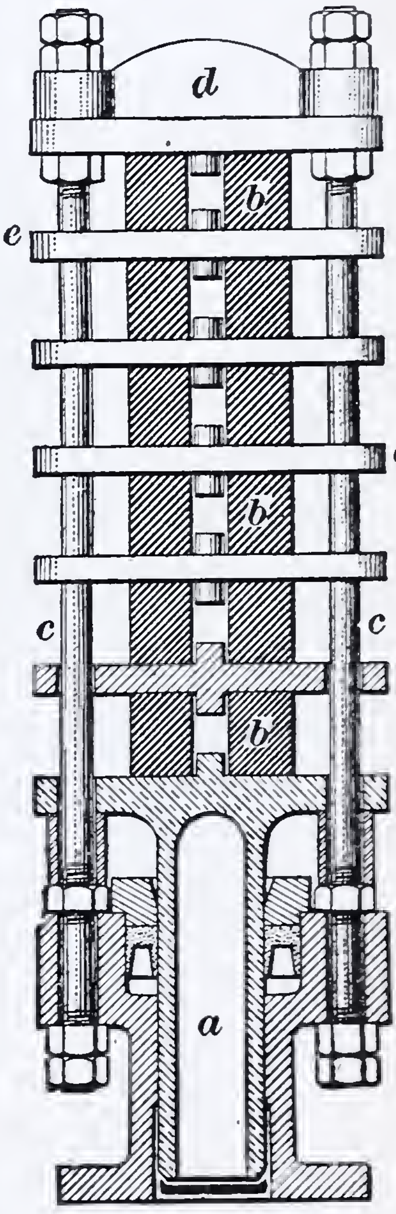


Fig. 45

74. Suction-Pipe Air Chamber.—With a long suction pipe or one with bends and valves, the resistance to the flow of the water will be considerable, and force will be required to start and stop the water with each stroke of the pump. In some cases, the force required is so great that the pressure of the atmosphere is not sufficient to set the column of water in motion quickly enough to fill the pump chamber as fast as the piston moves. This makes the action of the pump imperfect and causes a severe blow, called the *water hammer*, or *pounding*, when the piston again meets the inflowing water. This difficulty can be remedied by the use of an air chamber, usually called a *vacuum chamber* or *suction air chamber*, attached to the suction pipe near the pump. In its general form, this resembles an air chamber on the delivery pipe, but the pressure in it instead of being greater is always less than the atmospheric pressure. When the pump is drawing water, the air in the suction chamber expands and forces the water below it into the pump; at the same time, the pressure of the atmosphere forces water in through the suction pipe to balance the reduced pressure in the suction chamber. The suction chamber is again

partly filled and the air in it is compressed during the discharge stroke of the pump. It thus acts as a reservoir that receives from the suction pipe a nearly steady supply of water, which is given up intermittently to the pump.

For ordinary cases, the vacuum chamber may be made half the size of an air chamber working under the same conditions. A good rule is to make the cubic capacity of the vacuum chamber for a single pump twice that of the displacement of the piston for a single stroke.

STEAM PUMPS

(PART 2)

Serial 3029B Edition 1

DIRECT-ACTING STEAM PUMPS—(Continued)

INSTALLATION OF STEAM PUMPS

FOUNDATIONS

1. General Considerations.—The foundation for pumping machinery depends entirely on the type of pump. Direct-acting duplex pumps probably require the least foundation, for in them the piston and plunger on one side are moving in the opposite direction to that of the piston and plunger on the other side, with the result that the balancing of the machine in line with the plunger motion is almost complete and the strains due to the reversal of the motion of the moving parts on each stroke are contained almost wholly within the machine itself. Small duplex-pump foundations are made of a solid mass of brick, or concrete, whereas large pumps are often set on separate piers, one for the water ends and one for each pair of steam ends if the pump is compound or triple expansion. The foundation must go down to sufficiently firm soil to sustain the weight of the pump; or, if the soil is loose sand gravel, the foundation must be spread so that the pressure does not exceed, say, 1 ton per square foot. The minimum depth for a small pump, regardless of the nature of the soil, should not be less than 2 feet, so that there will be sufficient earth around the foundation to prevent it moving laterally.

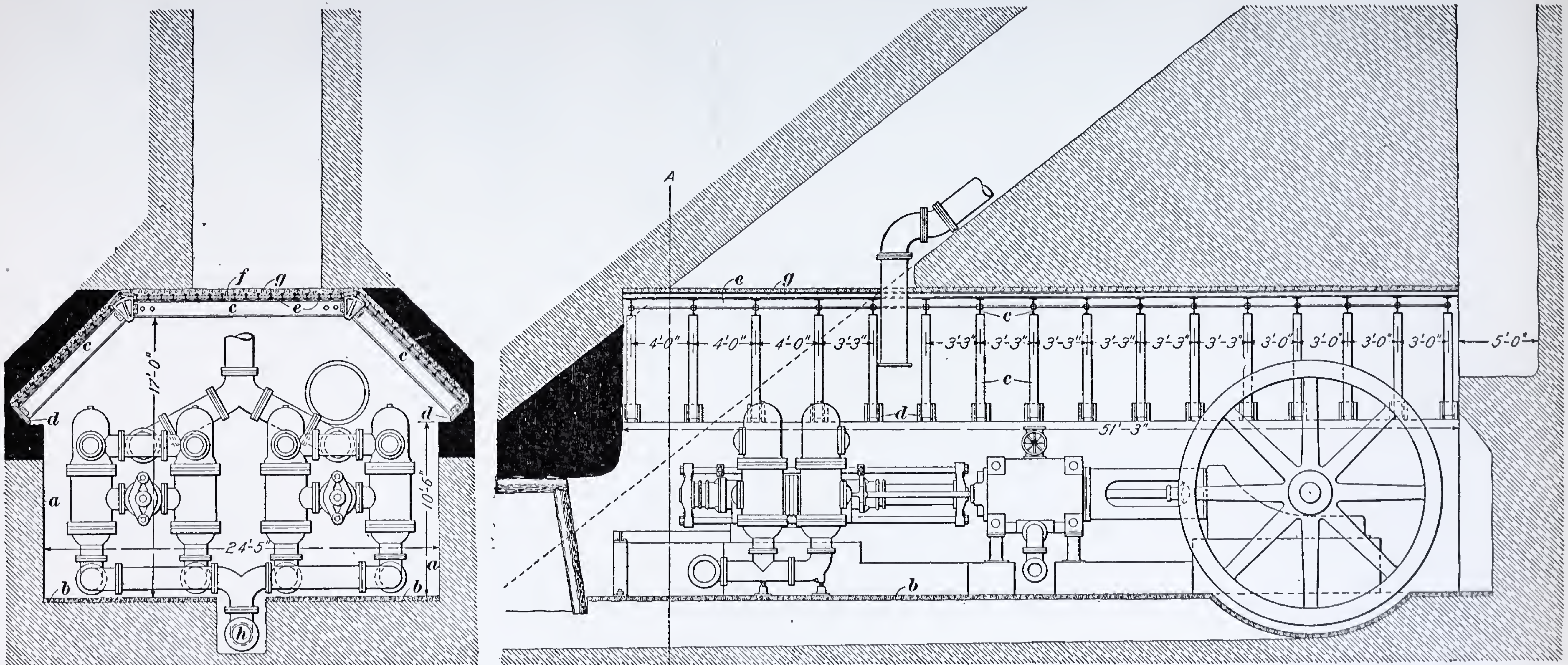
2. Setting Pump.—Foundations should be built of hard brick laid in cement mortar, or of concrete, or, in the case of

large pumps, of stone, if it can be readily secured. All pumps should be held down by foundation bolts. In the case of small pumps, the bolts are provided with a steel or wrought-iron plate washer built solidly into the foundation, while large pumps have pockets in the foundation for access to the lower foundation washer and nut. If the foundation bolts are built in solid, box washers should be used.

Underground mine pumps can dispense with foundations if solid rock exists at the place where the pump is to be located. All that is then required is to level off the surface of the rock to suit the bedplate or footings of the pump, and to set it directly on the rock. Holes for the foundation bolts are then drilled to a sufficient depth and the bolts, which should have a good length and a roughened shank, are fastened in place by pouring molten lead around them in the hole.

A foundation templet should always be used in which the foundation-bolt holes are carefully laid off, preferably from the actual castings, and the various heights of bosses or thicknesses of casting through which the bolts pass are marked. The templet should be carefully set with reference to the suction and delivery connections, so that when the pump is set up, the fittings and pipes will connect up properly. In large pumps it is customary to arrange the pipe connections in such a way that a short space is left between the piping and the pump. This space is then measured after the pump and piping are in place, and a distance piece is made to suit the measurement and then put in place.

3. Small pumps of the single-cylinder and duplex type are usually provided with two points of support only, one of which is rigidly bolted to the foundation, while the other is left free. This prevents the pump being thrown out of line, if properly constructed originally. When both the steam and water ends are bolted down, care must be taken not to twist or throw the pump out of line. In making the steam and water connections, the pipes should come fair to their connections and should not be sprung into place. Stresses on the pump structure due to winding foundation surfaces and sprung pipe



Section at A-B.

Fig. 1

connections should be guarded against, particularly with steam-thrown valves, as these are very sensitive and must be perfectly free. Any slight springing of the valve chamber will bind the valve and prevent its operating.

4. Pump Stations.—Pump houses, or pump stations as they are usually called, are generally placed near the foot of a shaft or slope and generally in a room excavated from the solid rock or in the coal. The roof may not need support or it may be necessary to support it by timbers or by steel supports. Fig. 1 shows a pump house of the Lehigh Valley Coal Com-

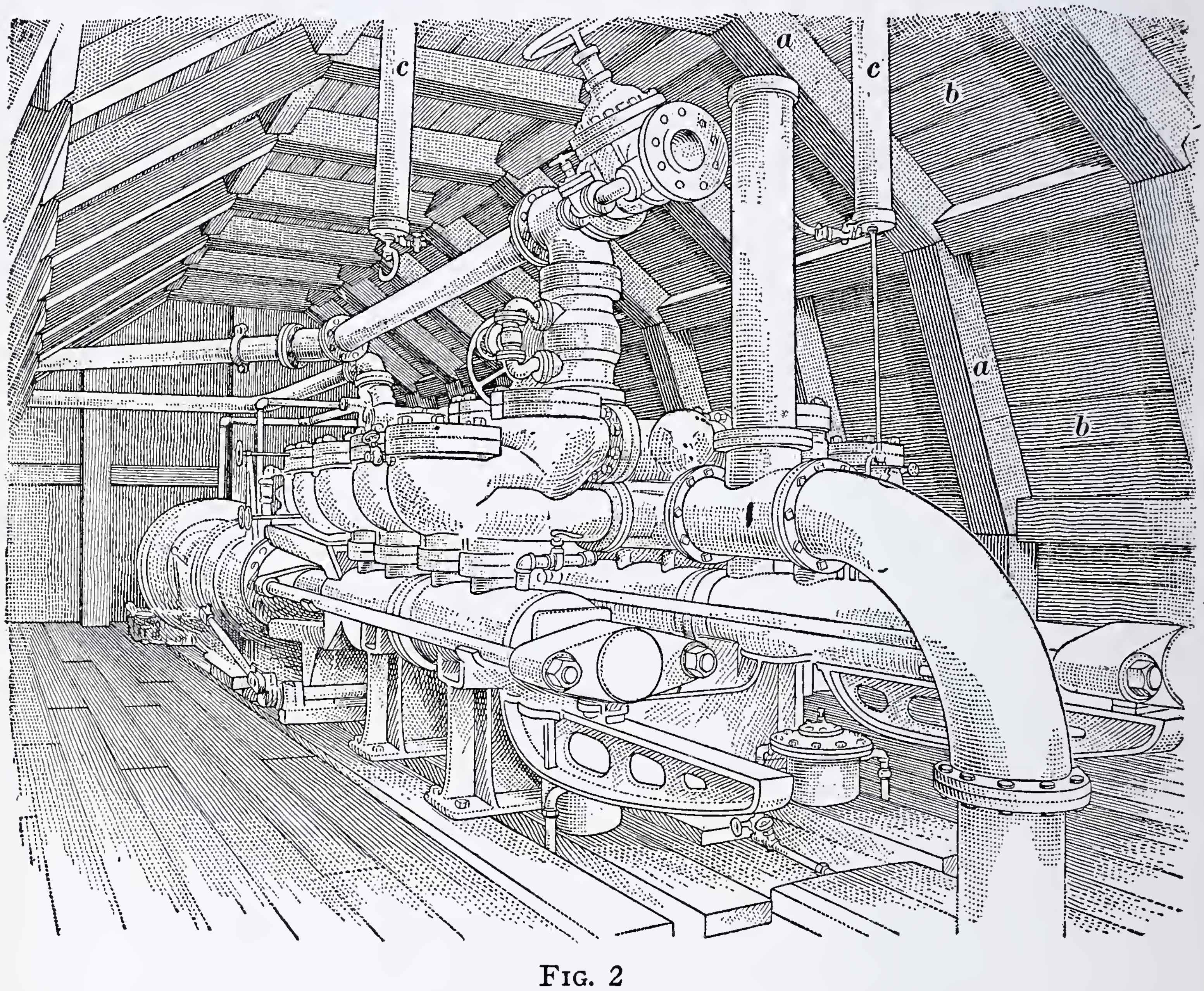


FIG. 2

pany, at Hazleton, Pennsylvania. The excavation for the house was made partly in the coal bed and partly in the underlying rock. The walls *a* were roughly plastered, the floor *b* was of concrete 3 inches thick and the roof was built of **I** beams *c* resting on cast-iron base plates *d* laid on the rock ledge, as shown. Steel rails *e* were laid on top of the **I** beams and the spaces between these rails were filled with brick *f*. Above the brick, concrete *g* was rammed. The suction pipe *h*

of the pump was laid in a trench below the floor level and leads thence to the sump. Fig. 2 shows a pump station in which the roof and sides are supported by heavy timber framing *a* backed by plank lagging *b*. The station is also provided with two compressed-air hoists *c* for use in handling pump parts when it is necessary to make repairs.

SUCTION PIPING

5. Requirements of Suction Pipe.—It is a good plan to have the suction pipe slightly larger than the discharge pipe. Wrought-iron pipe from 6 to 12 inches in diameter with screw flanges may be used for suction pipes to advantage; but above this size cast-iron flanged pipes are used, since wrought-iron pipe of this diameter is difficult to obtain, and if of sufficient strength is more expensive. The kind of pipe to be used will depend on the mine water; bronze pipes, lead- or wooden-lined iron pipes may be required if the water is corrosive for iron or steel.

Suction pipe lines should have the same diameter from end to end, since reductions disturb the uniform flow of water; they should be straight, but if elbows are necessary they should have a large radius and not be right angles. The diameter of the pipe should be such that the velocity of the water due to suction does not exceed 200 feet per minute. When the lift is vertical, it should not be over 18 feet without a suction air chamber being supplied, in order to add uniformity to the flow of water.

When suction pipes have a long horizontal run, they should be one or two sizes larger than the pump would require under ordinary conditions and be connected to the pump by a short funnel-shaped reducer. Pumps are frequently placed near the shaft, and draw their water from a distance in almost horizontal suction pipes; this is probably the most trying position for a pump and the engineer, since the pump will work best when near the source of water supply.

6. Foot-Valves.—A foot-valve is a check-valve placed at the lower end of the suction pipe below the water level in the

source of supply and opening toward the pump. Its purpose is to prevent the suction pipe emptying while the pump is at rest and to prevent the water in the suction pipe slipping back while running. When the water flows to the pump by gravity, a foot-valve is superfluous; but when the water is lifted by suction it is often fitted, since it will insure a prompt starting of the pump, providing that it is tight enough to hold the water in the suction pipe. In very cold weather and in exposed locations, the foot-valve constitutes an element of danger when the pump is out of use, since it prevents the emptying of the suction pipe The water in the latter may freeze and burst the pipe. To prevent this, a drain may advantageously be fitted to the lower end of the suction pipe, which is used in cold weather to empty the pipe if the pump is to stand idle for a long time.

7. When foot-valves are used, a relief valve may advantageously be placed on the suction pipe. Generally, the suction pipe is made considerably lighter than other parts of the pump, and if the suction valves should leak when the pump is standing or if the priming pipe be left open, the full pressure of the delivery water will come on the suction pipe and foot-valve, which are not usually designed to withstand such pressures. The relief valve, which should be set to relieve the pipe at a pressure well within its safe strength, prevents overstraining of the suction pipe from this cause. Foot-valves should be chosen with the greatest care; they should be simple and, preferably, of the weighted-lift type or clack valve, and should have at least 50 per cent. excess of area over the suction pipe.

8. Strainers.—When much timbering is done in a mine, small chips will find their way into the sump and, if not prevented, will be drawn into the pump and possibly lodge under a valve. Small stones and pieces of coal will also find their way into the pump if the sump becomes filled with dirt up to or within the suction zone of the tail-pipe. When such substances lodge under a valve, they prevent its proper seating and also prevent the pump from drawing, or discharging, its full quantity of water. To prevent the entrance of foreign

substances into the pump, some form of metal strainer is fitted to the suction pipe.

One kind of strainer, called a *basket strainer*, or *snorer*, shown in Fig. 3, may be fitted to the end of the suction pipe below the foot-valve. The round holes for the entrance of the water must be sufficient in number and size to have at least 50 per cent. more area than the suction pipe. Sometimes, to prevent the larger pieces of foreign material from getting to the strainer, the end of the suction pipe is placed inside a large perforated wooden box; or, the perforated box may be used alone and without the strainer.

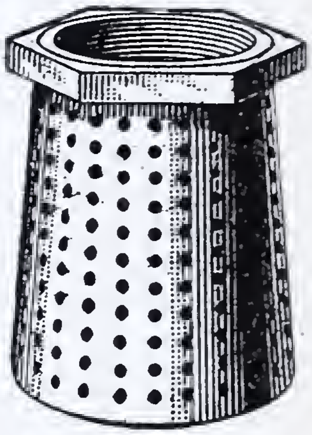


Fig. 3

Another form of strainer consists of a chamber placed in the suction pipe in an accessible position above the level of the water in the sump, and containing strainer plates provided with straight openings, or slots, like the spaces between grate bars. This strainer is sometimes connected directly with the pump, but it should not be so placed as to interfere with the removal of the heads from the water cylinder; and a short piece of pipe placed between the strainer and the pump will prevent this interference. The strainer plates are arranged so that they may be easily removed for cleaning, and in this feature possess an advantage over strainers placed at the end of the tail-pipe.

DISCHARGE PIPING

9. Requirements of Discharge Pipe.—The discharge pipe should be a plain, straight pipe from pump to terminal; but if bends are necessary they should be of as long a radius as possible. The discharge pipes at most mines are of cast iron, made in lengths that vary from 4 to 12 feet, according to their diameters. Pipes of small diameters are not made as heavy as pipes of large diameters, hence their length may be greater. The flanges cast to each end of the pipe are planed for even gasket surfaces and are drilled for bolts. Heavy pipes on inclines are made somewhat longer than those used in shafts, for the reasons that they can be bolted together and lined up more easily than when hanging in shafts.

Since the tendency to rupture will increase with the head and the diameter of the pipe, it follows that the greater the head of water and the diameter of the pipe the more metal will be required to withstand the pressure. Experiments with cast-iron pipes have demonstrated that uniform castings cannot be made, and where the metal has been less than $\frac{1}{2}$ inch thick no dependence could be placed on it on account of blow-holes.

10. Gate Valve.—In case the pump column is filled with water and the pump is stopped, the water will run back through the pump if the foot-valve is not tight. To prevent this, a gate or check-valve, preferably the former, on account of its wearing less, is placed a short distance from the pump in the column pipe. A gate valve, while useful in the column pipe to keep the pressure off the valves when the pump is not at work, is also useful for keeping water from running back into the pump chamber when the valves are being repaired. Some use check-valves in the column pipe, but they are objectionable in that they increase friction and are liable to become broken.

11. Air-Discharge Valves.—When a check-valve is not used in the delivery pipe and the space between the suction and delivery valves is large and the delivery pipe is full of water, the pump will often refuse to start the water in the suction end, owing to compressed air being trapped between the water in the delivery deck and suction valves. Air-discharge valves, each composed of a globe valve and a check-valve, may then be used on each head, the check-valve opening to the atmosphere, thus permitting the escape of air but preventing its entrance when the globe valve is open. The globe valve is closed when the pump is working properly, as shown by water coming from the check-valve.

Violent jarring and trembling often take place if the discharge chamber is not provided with an air chamber where the lift is not above 150 feet, and for lifts above that distance, when there is no alleviator. This is due to the column of water in the discharge pipe coming to rest suddenly between strokes and again meeting the power that must put it in motion again.

12. Velocity of Flow.—The velocity of the water flowing through the delivery pipe for direct-acting pumps should not exceed 400 feet per minute, while for large crank-and-fly-wheel pumping engines the velocity of water in both suction and delivery pipes is about 300 feet per minute. If the suction pipe is made small, the pump will fail to fill and the plunger will strike the incoming water on its return stroke, producing a violent and dangerous shock. If the delivery pipe is made small the cost of power required to force the water through the pipes at a high velocity will very quickly overrun the interest and depreciation on a larger pipe.

13. Acid-Resisting Materials.—In case the water is acid, cast-iron or wrought-iron pipes are much corroded and eaten away, to prevent which they are often lined with wooden staves, as shown at *a*, Fig. 4. At the Parrott Mines in Montana, the pipes are made of 1-inch bronze, 5 inches inside diameter and 12 feet long. Cast-iron pipes lined with sheet lead are also often used to resist the corrosion of acid mine water. Some mine waters contain as much as 160 grains of sulphuric and other acids per gallon of water, and often the most difficult part of a pumping proposition is the securing of a material for the piping and pumps that will withstand the corrosion. Various acid-resisting alloys are used for this purpose.

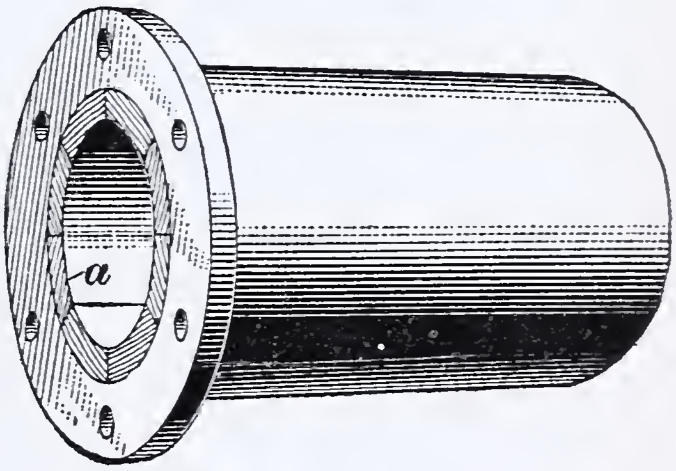


Fig. 4

Not only must the piping be protected from the corrosive water but often, as well, all parts of the pump coming in contact with the water, thus frequently greatly increasing the cost of the pumping equipment.

Mine waters also carry in solution impurities that will sometimes be deposited in the discharge pipes to such an extent as to nearly close them. Usually this is sulphate of lime or aluminum, and has the effect of making the pump labor without accomplishing its duty. There seems to be no remedy for this except changing the pipes.

AUXILIARY PIPING

14. Priming Pipe.—The priming, or charging, pipe is a small pipe connecting the discharge pipe from a point above the gate valve with the suction chamber of the pump. It is particularly useful in the case of long suction lifts to fill the working chamber and suction pipe with water, taking up all clearances and helping the pump to take hold of the water quickly. This pipe may be from $\frac{3}{4}$ of 1 per cent. to 1 per cent. of the area of the plunger; its size is a matter of little importance, but it should be large enough to fill the suction pipe and pump chamber in a reasonable time, which will depend somewhat on the size and design of the pump chamber and the length of suction pipe. A pipe much larger than 1 per cent. of the plunger area will be required in the case of long inclined or horizontal suction pipes.

15. Waste Pipe.—A waste, or starting, pipe that can be led into any convenient place of overflow should be provided so that the pump, in starting, can free itself of air. It often happens that a pump refuses to lift while the full pressure against which it is expected to work is resting on the discharge valves, for the reason that the air within the pump chamber is not dislodged, but only compressed and expanded by the motion of the plunger. A pump in this condition is said to be *air-bound.* It is customary, in this case, to discharge the water and air from the pump through a waste pipe fitted with a valve and connected to the discharge pipe close to the pump. When the air is all out of the pump, the valve in the waste pipe is closed and the pump is in condition to discharge water into the column pipe.

Proper drain pipes and drain valves should be provided for all parts of the pump and the pipe connections, and, in short, for all parts in which water may remain when the pump is not in use, and give trouble by freezing.

16. By-Pass Pipes.—By-pass pipes are usually attached to the water end of compound pumps, so that they will connect the discharge and suction chambers and permit the water to be pumped over and over again and prevent accidents that would

arise in case the pump ran out of water. When starting compound pumps, the steam on the high-pressure-cylinder piston is not always powerful enough to move the plungers against the resistance of the water in the discharge pipe; but, by opening the gate valve in the by-pass piping, the pressure on the plunger is relieved for a sufficient number of strokes to allow the steam to reach the low-pressure piston, when the combined force of the two pistons will do the work and the by-pass pipe can be closed.

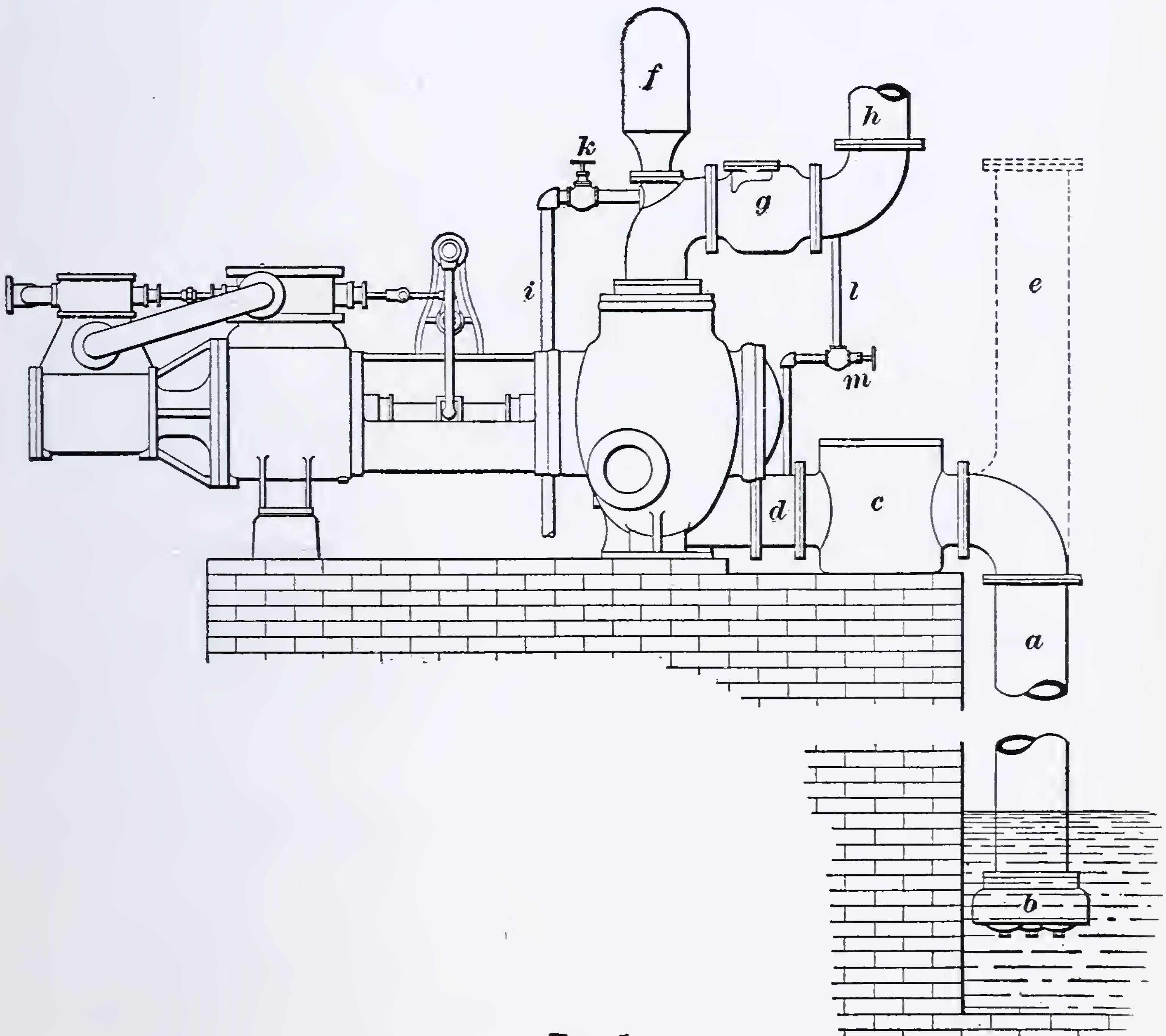


Fig. 5

Steam-end by-pass pipes are generally fitted to compound and triple-expansion pumps for the purpose of warming up the intermediate and low-pressure cylinders before starting the pump. They are provided with valves and take high-pressure steam either directly from the steam main or from the high-pressure valve chest.

17. General Arrangement of Piping.—The general arrangement of a pump and its piping is shown in Fig. 5. The suction

pipe *a* is fitted with the foot-valve *b* and has a strainer *c* placed close to the pump, from which it is separated by the short distance piece *d*. With a vertical suction pipe, the suction air chamber may be placed as shown by the dotted lines at *e*. An air chamber *f* is placed on the column pipe between the check-valve *g* and the pump discharge valves. The waste pipe *i* is connected to the discharge pipe between the pump discharge valves and the check-valve *g*. It is fitted with the valve *k*. The discharge pipe *h* is connected to the suction pipe close to the pump by the priming pipe *l*, which is fitted with the valve *m*.

MANAGEMENT OF STEAM PUMPS

INTRODUCTION

18. Care of Pumps.—To care for a pump properly, one should understand its mechanism; and if the pump man has never previously had charge of a similar pump, he should have at hand a detailed drawing of its parts. Only capable men should be placed in charge of mine pumps, since the work of changing valves, packing the pistons or plungers, and adjusting nuts and bolts with a monkeywrench requires as much intelligence as many other matters that enter into pump running. The influence that the atmospheric pressure exerts on pump suction, the friction of valves and column pipe, the causes for valves not working properly, the relation existing between the steam end and water end, and the available steam pressure and water head are some of the matters that have an important bearing on the working of a pump, and this knowledge a pump runner should possess in order to be capable.

All, pumps, when new, should be run slowly until the parts have become thoroughly adjusted to their bearings, when the speed may be increased. Because a new pump works stiffly is no cause for alarm, for, while a machinist can properly construct the parts, he cannot always foresee the strains caused by the action of the pump when the parts are assembled, and which require certain adjustments to be made after the pump

is at work. By running the pump slowly with the parts properly lubricated and making such adjustments as may be necessary, stiffness will gradually disappear and the pump's highest efficiency will be attained, provided that other matters on which the pump's action depends have received proper attention.

19. Losses in Pump Efficiency.—The causes that affect a pump, impair its efficiency, and prevent its performing its full duty are: wear; the improper adjustment of the valves, valve stems and levers; the improper packing of the plungers and stuffingboxes; drawing up the stuffingbox glands too tightly; lost motion due to permitting the working parts to wear and not adjusting them to the new condition; accumulations of foreign matter under the valves or in the strainer; broken valves and valve springs; leakage in valves; taking air in the suction pipe; finally, clogged or broken discharge pipes, and neglecting to use proper gaskets wherever required.

At many mines, the pumps are capable of a larger capacity than is obtained by the speed at which they are run, but it may nevertheless be important that such pumps should be run continually, as any serious interruption in pumping might cause trouble elsewhere. It is customary, therefore, to keep on hand a supply of duplicate valves, moving parts, and packing, in order that when it becomes necessary to make repairs this may be done without great loss of time.

CARE DURING OPERATION

20. Cleaning Pipes and Cylinders.—The first step after a pump has been erected is to clean out the steam piping. In order that this may be done without carrying foreign matter into the pump, the piping is left disconnected from the pump, and steam at full boiler pressure is allowed to blow freely through the piping and valves for a few minutes. Steam is then shut off and the piping connected to the pump.

The next step is to blow out the steam cylinders. To do this, the cylinder heads should be put on, leaving the pistons and

valves out of the cylinders. The stuffingboxes should be closed, which is most conveniently done by placing a piece of board between the stuffingbox and the reversed gland and then setting up the nuts on the stuffingbox studs. When the gland is drawn home by a nut outside of it, a circular piece of pine board may be placed between the end of the gland and the inside of the nut in order to close the opening through which the piston rod passes. The steam may now be turned on the main steam pipe leading to the pump; by opening the throttle valve wide at short intervals, the sand and scale in the ports and other passages and spaces of the steam end can be blown out. After the cylinders have been blown out, the heads and covers should be removed, and all foreign matter blown into the corners and chambers of the cylinders removed by hand. The pistons, valves, cylinder heads, and other covers can now be put in place. The blowing out of the pipes and cylinders after erection is often neglected or but imperfectly done, with serious consequences to the machine; it cannot be too thoroughly done, particularly in that type of pump where the steam ports and exhaust ports are on top, for in this particular case the sand and grit are deposited in the bottom of the cylinder for the piston to ride on.

21. Packing.—The packing of all rods and stems is the next step. If fibrous packing is used, the boxes should be filled full and the glands tightened down very moderately. The tightening of the glands can best be done when steam is on and the machine is in motion, when they should be tightened only sufficiently to stop leakage and no more. When excessive tightening is required to stop leakage, the packing should be completely renewed. Some pumps are fitted with metallic packing; this packing is usually fitted up by specialists who fully guarantee it and their directions for use should be carefully followed; in case of failure or unsatisfactory results, the makers should be consulted.

22. Oiling.—The oiling of the machinery is the next step and is a very important one. All rubbing surfaces should be provided with suitable oiling devices appropriate to the particu-

lar place and service. The quality of oil should be carefully selected to suit the velocity and pressure of the rubbing surfaces on which it is used. For use within the steam cylinder, heavy mineral oil is the only oil capable of withstanding the high temperature; and in starting up new pumps only the best quality should be used, regardless of price. A liberal use of this oil for the first month will go far toward reducing subsequent oil bills.

The pumping engine must often run continuously and without interruption for a month or even longer. This requires that all oiling devices be so arranged that they can be supplied and adjusted while the machine is in motion. It is a good plan to provide two sets of oiling systems for all the principal journals, so that if one fails the other can be used while the disabled one is being overhauled. All oil holes should have been filled with wooden plugs, bits of waste twisted into the hole, or some other protection, while the machine was being erected. These should now all be removed and all oil holes and oil channels thoroughly cleaned out. Bearings should be flooded with oil at first to wash out any dust or grit that may have reached the rubbing surfaces.

23. Preparing Steam and Water Ends.—The steam end is now ready to be warmed up. Here it may be mentioned that from now on the method of starting a pump is the same whether the pump is a new one or an old one. To warm up the steam end, the throttle is opened just a little, the drain cocks are opened wide and steam is allowed to blow through the cylinder until no more water comes out of the drain cocks, using the steam by-pass pipes in case of multiple expansion pumps. If the pump has a valve gear that can be operated by hand, the warming up can be hastened by working the valve back and forth slowly. While the steam end is warming up, the water end should be got ready by opening the stop-valve in the delivery pipe, and otherwise seeing to it that the pump has a free delivery. If a stop-valve is fitted to the suction pipe, this should be opened. If the machine is compound or triple expansion, the water by-pass valves must be opened

until the machine has made a sufficient number of strokes to bring the intermediate and low-pressure cylinders into action, when the by-pass valves should be closed. If the pump is fitted with dash-relief valves, these should be closed before starting in order to keep the pistons as far from the heads as possible in starting. Should the pump exhaust into an independent condenser, this should be started and a vacuum obtained before starting the pumps.

24. Starting and Stopping.—To start the pump, the throttle is opened slowly and the pistons allowed to work back and forth very slowly a few times, gradually increasing in velocity until they attain their full speed. After the pump has been running a couple of minutes, the drain cocks are closed. If the pump has dash-relief valves, the length of stroke may now be carefully adjusted.

To stop the pump, the throttle is closed, the drain cocks opened, and the gate valve in the discharge pipe, if one is fitted, is closed. The condenser is shut down.

PUMP TROUBLES

25. Suction-End Troubles.—A common reason why pumps refuse to work properly is that they take air below the suction valves. Small leaks will cause the piston to jump, as the water will not enter through the suction valves soon enough to fill the entire chamber. This trouble may be remedied by seeing that all joints in the suction pipe and between the pipe and the pump are air-tight. Leaks may sometimes be detected by the hearing, or by the flame from a candle being drawn toward the hole. If the leaks are small and not at the pipe joints, a coat of asphalt paint may stop them; if large, they should be drilled larger, the hole threaded, and a screw plug inserted. If the leak is at the joint between two pipes, the pipes should be uncoupled and screwed together again, using graphite pipe grease for a lubricant; or, if the joint is a flanged one, a new gasket should be placed between the flanges, and the pipes lined up, before the bolts are tightened.

Sometimes leaky valves in the suction chamber prevent the pump from doing its full duty. This may be caused by corrosion of the valve and valve seat; by chips or gravel under the valves, preventing their seating properly; or the valves and seats may have become so worn that leakage cannot be prevented without changing the parts. If a suction strainer or the end of the suction pipe becomes embedded in sand or clogged with foreign matter, the supply will be cut off from a pump.

26. If the pump pounds soon after the beginning of a stroke, when running fast, it shows that the pump chamber is not filling and that the plunger is striking the incoming water on its return stroke. A suction air chamber will help to remedy the evil. Pounding in pumps is sometimes caused by the water lagging behind the plunger, due to the friction of a small, long, horizontal suction pipe. When suction pipes have a long horizontal run, they should be one or two sizes larger than is required for capacity alone.

27. Air pockets under the delivery valve deck, caused either by bad design or a shifting core, will very much reduce the capacity and efficiency of a pump. The effect of the air pocket is to entrap air, which is compressed to delivery-water pressure and expands again on the suction stroke. If the relative capacity of the pocket to the plunger displacement is sufficient, the entrapped air will expand to atmospheric pressure, reducing the suction lift to zero. This defect, however small, will always reduce the suction lift and is not easy to remedy; its existence should always be cause for the rejection of a pump. The suction pipe should always be tested for leaks before it is covered, if laid in a trench or otherwise made inaccessible, because it must be made tight before the pump will work successfully.

28. Water-End Troubles.—Leaks in the delivery pipe, while common and at times more difficult to remedy than leaks in the suction, are plainly evident. They do not affect the action of the pump or its efficiency to any extent, the

loss being exactly proportional to the magnitude of the leak. Small pinholes in delivery pipes can often be stopped up by the same means given in Art. **25** for suction pipes. If the leak is extensive, however, it will generally be necessary to use a *pipe clamp*. Such clamps may be made in a good many different ways, according to the location and extent of the leak and the facilities for repair. One of the simplest pipe clamps is shown in Fig. 6. It consists simply of a piece of sheet iron or sheet steel of sufficient width to cover the leak and bent to the form shown. A piece of sheet packing, which may be covered with red-lead putty to advantage, is placed over the leak and the pipe clamp is then placed over this and the ends drawn together by the bolt shown. The clamp shown is adapted for small pipes only. For large pipes the clamp must be made in halves.

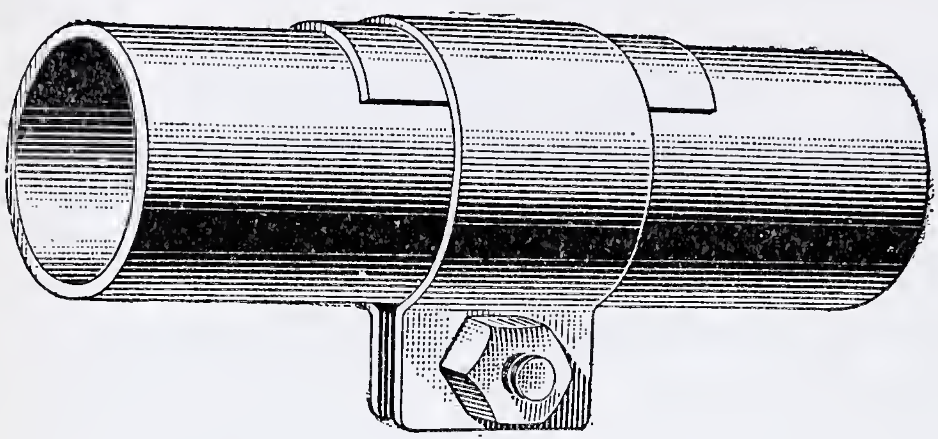


FIG. 6

Violent jarring and trembling of the pump arises when the delivery air chamber becomes filled with water. It should be recharged with air by means of the air-charging pump, a near-by air compressor, or by a hand air pump.

29. Leakage of Pistons and Plungers.—With piston pumps and inside-packed plunger pumps there is likely to be considerable unnoticed leakage, so that the water passes from one side of the piston or plunger to the other. If this leakage is extensive, it can be heard by placing the ear against the pump chamber. With these types of pump it is best to make regular inspections for leakage past the piston or plunger, the manner of testing depending on the design of the pump. The general method of testing is to subject one side of the plunger to an air pressure or a hydrostatic pressure at least equal to the working pressure, while the other side of the plunger is exposed for inspection. If leaks are discovered, judgment must be used as to the manner of repairing them or whether to condemn

the plunger. With outside-packed plungers there can be no unobserved leaks past them, and this is one of the principal reasons for their use.

30. Leakage of Valves.—Leaks past the suction and delivery valves can readily be tested when the piston or plunger is being tested for leaks past it. The delivery and suction valves should be tested separately; the fact that the column of water in the delivery pipe does not drain out while standing is not proof that both sets of valves are tight, since either set will support the water while the other set may be leaking badly. To test the suction valves for leakage, disconnect the suction pipe or take any other convenient steps that will allow the leakage to be seen. Fill the delivery pipe full of water, having removed enough delivery valves to allow the pressure to reach all the suction valves, and observe which valves, if any, are leaking. When there is a valve in the delivery pipe, this may be shut and water pumped into the pump cylinder with a small force pump, running the pressure up to the working pressure. Care must be taken, by removing delivery valves if necessary, that the pressure reaches all the suction valves.

The delivery valves can be tested by filling the delivery pipe or by closing the valve in the delivery pipe and pumping water into the delivery pipe between its valve and the pump delivery valve. The pump chamber must be open so that the leaks can be seen.

31. Steam-End Troubles.—Valves in the steam end sometimes wear unevenly or the valve stems by continual action wear and cause lost motion, thus causing a back pressure and irregular action. Anything wrong in the steam end can be determined by the irregular exhaust usually, but even this may be deceptive in case the water end valves are leaking. If the steam valves are suspected, the cover to the steam chest is raised for their inspection, but the valves are not to be disturbed until it has been proved that they do not open and close properly by moving the water piston backwards and forwards several times. The seat of the trouble may then

be found to exist not in the valves but in the levers or toggles that throw them, and the adjustments may be properly made without disturbing the valves.

32. Steam-Valve Leakage.—Sometimes there are soft spots in the valve seat or valve that wear faster than the rest of the valve and seat. Through these slight depressions, steam will blow and cut both valve and seat if attention is not given them; back pressure will then seriously interfere with the working of the pump. If the defect is in the valve, a new one can take its place; but the valve seat, if a part of the steam cylinder, will require an entire new cylinder, and hence it is economy to scrape the seat until the depressions are removed. A try plate made of steel having a perfectly level surface is covered with chalk and carefully rubbed over the valve seat. The elevations will have chalk on them, the depressions will not. The elevations are scraped with a chisel made of the best steel until they are worn down so that chalk sticks to every part of the seat alike. The valve is treated in the same way if it can be done without too much expense. The valve and the valve seat when removable should be sent to the shop to be reground.

33. Lost Motion in Duplex Pumps.—In many duplex pumps, there are very slight differences between the two sides, and the amount of lost motion between the valve stem and the valve should be carefully adjusted. Too little lost motion will cause short stroking, while too much will allow the pistons to strike the heads. It becomes a very delicate matter and one requiring judgment to make proper adjustments for lost motion. In case new parts are substituted for worn or broken parts, the pump must be run slowly until, by careful adjustment, the timing for cut-off and admission of steam are perfectly balanced.

STEAM-PUMP CALCULATIONS

INTRODUCTION

34. Pump Sizes.—The size of a direct-acting steam pump is commonly expressed by giving, in the order named, the diameter of the steam cylinder, the diameter of the water piston or plunger, and the common length of stroke of the water and steam ends, all dimensions being given in inches. A $10'' \times 6'' \times 12''$ pump is, then, a single pump with a 10-inch steam piston, a 6-inch plunger, and a stroke of 12 inches. In the case of duplex pumps, the dimensions of one side only are given, the word *duplex* being inserted in the description; thus, $12'' \times 8'' \times 16''$ duplex pump. A pump described as a $10''$ and $20'' \times 9'' \times 18''$ compound duplex pump is made up of two similar units, each of which consists of a high-pressure cylinder 10 inches in diameter, a low-pressure cylinder 20 inches in diameter, and a plunger 9 inches in diameter, the moving parts having a common stroke of 18 inches.

35. Conversion Factors.—As the discharge of a pump may be given in cubic feet, gallons, or pounds of water, it is necessary to be able to convert one unit quickly into either of the others. The following conversion factors are all that are necessary for this purpose: One cubic foot of water contains 1,728 cubic inches, or 7.48 U. S. gallons of 231 cubic inches, and weighs 62.5 pounds. One gallon weighs 8.35 pounds. The Imperial gallon, of 277.42 cubic inches and weighing 10 pounds, is used in Canada.

When the head against which a pump is working is given in feet, the equivalent pressure, in pounds per square inch, may be found by multiplying the head by .434, which is the weight of a column of water 1 foot high and 1 square inch in cross-section. When the pressure against which the pump is working is given in pounds per square inch, the equivalent head, in feet, may be found by multiplying the pressure by 2.304.

36. Flow of Water Into Mines.—Before it is possible to calculate the size of pump required to drain a mine, it is

necessary to know the quantity of water entering the mine. The most accurate way to determine the rate of inflow is to note the time it takes to fill a sump of known dimensions to a measured depth. The amount of water entering a mine varies greatly with the season of the year, the geological structure of the region, and the nature of the mining operations; but it is advisable to design the pumping plant to handle the maximum inflow, and to reduce the speed of the pump or the number of hours it is in operation as the inflow decreases.

EXAMPLE.—A mine pump having been closed down for repairs, the sump, which is 60 feet long and 10 feet wide, filled to a depth of 4 feet in 4 hours, at the end of which time the pump was started. (*a*) How much water entered the mine in 4 hours, and (*b*) what is the rate of inflow in gallons per minute? (*c*) If the pump has a capacity of 125 gallons per minute, how long will it take it to drain the sump?

SOLUTION.—(*a*) The volume of water in the sump at the end of 4 hr. is $60 \times 10 \times 4 = 2{,}400$ cu. ft., or $2{,}400 \times 7.48 = 17{,}952$ gal. Ans.

(*b*) As the total inflow is 17,952 gal. in 4 hr. the inflow per minute is $17{,}952 \div (4 \times 60) = 74.8$, say 75 gal. Ans.

(*c*) As the pump has a capacity of 125 gal. per min. and the water flows into the sump at the rate of 75 gal. per min., the pump will gain on the water in the sump at the rate of $125 - 75 = 50$ gal. per min. As there is 17,952 gal. in the sump, the sump will be drained in $17{,}952 \div 50 = 359$ min., or about 6 hr. Ans.

37. Effect of Altitude on Suction Lift.—A common method of calculating the height to which a pump should draw water through its suction pipe is to multiply the atmospheric pressure by 2.304. Thus, the theoretical height of suction at sea level where the atmospheric pressure is 14.7 pounds per square inch, is $14.7 \times 2.304 = 33.87$ feet. As the atmospheric pressure decreases as the altitude increases, the height of suction decreases correspondingly. The atmospheric pressure at different altitudes is given in Table I, also the corresponding heights to which water can theoretically be drawn at these altitudes. The height of suction can also be calculated from the reading of the barometer. The specific gravity of mercury is 13.6; hence, the height of the water column, in feet, is equal to $13.6 \div 12 = 1.133$ times the barometer reading, in inches.

The theoretical height of suction is not obtainable in practice, because of friction in the pipes, leaky valves, etc., and the maximum suction lift possible at any altitude is about 80 per cent. of the theoretical height. Thus, at sea level the maximum lift will be about 28 feet; at 5,000 feet elevation the lift will be about 22.5 feet; and similarly at other elevations.

TABLE I

THEORETICAL HEIGHT OF SUCTION AT DIFFERENT ALTITUDES

Elevation Above Sea Level Feet	Atmospheric Pressure Pounds per Square Inch	Theoretical Height of Suction Feet	Elevation Above Sea Level Feet	Atmospheric Pressure Pounds per Square Inch	Theoretical Height of Suction Feet
Sea Level	14.70	33.87	8,000	11.02	25.39
500	14.44	33.27	8,500	10.82	24.93
1,000	14.18	32.67	9,000	10.62	24.47
1,500	13.93	32.09	9,500	10.44	24.06
2,000	13.68	31.52	10,000	10.25	23.62
2,500	13.43	30.94	10,500	10.07	23.20
3,000	13.19	30.39	11,000	9.89	22.79
3,500	12.96	29.86	11,500	9.71	22.37
4,000	12.73	29.33	12,000	9.54	21.98
4,500	12.50	28.80	12,500	9.37	21.59
5,000	12.27	28.27	13,000	9.20	21.20
5,500	12.06	27.78	13,500	9.04	20.83
6,000	11.84	27.28	14,000	8.87	20.44
6,500	11.63	26.79	14,500	8.72	20.09
7,000	11.42	26.31	15,000	8.56	19.72
7,500	11.22	25.85			

It must not be understood from the foregoing statements that, at sea level, the pump should be placed 28 feet above the sump. The effort is made to limit the height of suction to 16 or 18 feet, and the pump should be placed as close to the water as possible so as to reduce the lift, and, consequently, to reduce the work demanded of the pump.

Example 1.—If the mercury in the barometer reads 24 inches, how high can a pump draw water, assuming that the actual suction is 80 per cent. of the theoretical?

SOLUTION.—From the foregoing explanation, the height would be

$$24 \times 1.133 \times .80 = 21.75 \text{ ft.} \quad \text{Ans.}$$

EXAMPLE 2.—What is the theoretical height of suction at an elevation of 4,500 feet?

SOLUTION.—If a table showing the readings of the barometer at different elevations is not available, it may be assumed that the barometer falls 1 inch for each 900 ft. of elevation above sea level. Hence, at 4,500 ft. elevation, the barometer would read $30-(4{,}500 \div 900)=25$ in., and the theoretical height of suction would be approximately

$$25 \times 1.133 = 28.32 \text{ ft.} \quad \text{Ans.}$$

38. Effect of Temperature on Suction Lift.—At all temperatures above 32° F. water vaporizes, and this vapor exerts a back pressure, known as *vapor tension*, that increases with the temperature. The tension of the vapor opposes the atmospheric pressure and correspondingly reduces the height to which water will rise in the suction pipe. If the temperature of the water reaches 212° F., it cannot be drawn by suction into a pump at sea level, for the tension of the water vapor then equals the atmospheric pressure. Table II shows the theoretical height to which it is possible to raise water of different temperatures by suction, at sea level, when the barometer reads 30 inches.

TABLE II

INFLUENCE OF TEMPERATURE ON SUCTION LIFT, AT SEA LEVEL

Temperature Degrees Fahrenheit	Height of Lift Feet	Temperature Degrees Fahrenheit	Height of Lift Feet
32	33.7	130	29.2
40	33.6	140	27.7
50	33.5	150	25.9
60	33.4	160	23.5
70	33.1	170	20.6
80	32.9	180	17.1
90	32.5	190	12.8
100	32.0	200	7.6
110	31.3	210	1.4
120	30.3	212	0.0

WATER-END CALCULATIONS

39. Displacement.—The volume of water expelled from a pump cylinder by one stroke of the plunger, assuming that there is no slip, is termed the *pump displacement*, and is equal to the area of the plunger multiplied by the length of stroke. It is often assumed that the entire area of the plunger or piston is effectively employed in displacing water. An inspection of the illustrations in the preceding Section will show that this is true only in the case of outside-packed plunger pumps, in which the plunger protrudes through the end of the cylinder. With inside-packed and center-packed plunger pumps and double-acting piston pumps, in which the plunger or piston is attached to a rod, the effective area of one side of the piston is the total area of the piston, and the effective area of the other side is less than the total area by the area of the piston rod. The *mean effective area*, as it is called, of the piston is the average of the effective areas of the two sides of the piston. As shown in the following example, the deduction to be made for the area of the piston rod is not large and, in mining practice, is fully covered by the allowance made for the lack of efficiency in the water end. When an accurate knowledge of the displacement is required, the mean effective area must, of course, be used.

EXAMPLE.—A pump has a 10-inch piston, 20-inch stroke, and 2-inch piston rod. (*a*) What is the displacement when no allowance is made for the piston rod? (*b*) What is the actual displacement?

SOLUTION.—(*a*) The area of the piston, no allowance being made for the rod, is $10^2 \times .7854 = 78.54$ sq. in. The displacement is, then,

$$78.54 \times 20 = 1{,}570.8 \text{ cu. in. Ans.}$$

(*b*) The area of the piston rod is $2^2 \times .7854 = 3.14$ sq. in. and the effective area of the side of the piston to which it is attached is $78.54 - 3.14 = 75.40$ sq. in. The mean effective area of the piston is, then, $(78.54 + 75.40) \div 2 = 76.97$ sq. in. The actual displacement is

$$76.97 \times 20 = 1{,}539.4 \text{ cu. in. Ans.}$$

The actual displacement is 31.4 cu. in., or 2 per cent., less than the displacement when no allowance is made for the rod.

40. Slip.—The actual discharge of a pump is always less than the theoretical discharge because of leakage past the

valves and piston or plunger, and also because of the return of water through the valves while they are in the act of closing. The difference between the theoretical discharge and the actual discharge, expressed as a percentage of the theoretical discharge, is called the slip of the pump. The slip may be found by the formula

$$S = 100\ \frac{Q - Q'}{Q}$$

in which S = slip, in per cent.;
Q = theoretical discharge;
Q' = actual discharge.

EXAMPLE.—A single-acting plunger pump with a plunger 10 inches in diameter and 20-inch stroke discharges 33.5 cubic feet of water per minute when making 40 discharge strokes. What is the slip?

SOLUTION.—According to the example in Art. **39**, the displacement of this pump is 1,570.8 cu. in. In making 40 discharge strokes, the theoretical discharge is $1{,}570.8 \times 40 = 62{,}832$ cu. in., or $62{,}832 \div 1{,}728 = 36.36$ cu. ft. Since the actual discharge is $Q' = 33.5$ cu. ft., the slip is

$$S = 100 \times \frac{36.36 - 33.5}{36.36} = 7.87 \text{ per cent.} \quad \text{Ans.}$$

41. Efficiency of Water End.—The efficiency of the water end of a pump is equal to 100 per cent. minus the percentage of slip. Thus, in the case of the pump mentioned in the example of Art. **40**, the efficiency of the water end is $100 - 7.87 = 92.13$ per cent.

The slip in new pumps and in old pumps that are well cared for should not exceed 3 per cent., which means a water-end efficiency of 97 per cent. Excessive slip means that the pump is in poor condition and should be overhauled. It may be due to a worn plunger, an imperfectly packed piston, or to a worn, broken, or displaced valve caused by the breaking of a valve stem or the unfastening of a valve seat.

In the pumping examples sometimes given in state examinations for certificates of competency, it is quite common to be told to "make an allowance of 25 per cent. for slip," or "assume that the efficiency of the water end is 80 per cent." While it is undoubtedly true that pumps in service occasionally show such excessive slip, new pumps should not be designed

and installed with the intention that they will be neglected when in use; and neglect alone will account for high slippage.

42. Piston Speed.—The piston speed of a pump is the number of feet traveled per minute by the plunger when the pump is discharging water; that is, it equals the length of stroke in feet multiplied by the number of working, or discharge, strokes per minute. It should be most carefully noted that in single-acting pumps the number of discharge strokes is one-half the number of actual strokes, and that in double-acting pumps the number of discharge strokes equals the number of actual strokes. With duplex double-acting pumps it is customary, when giving the number of strokes, to refer to the number made by one plunger only, which is obviously but one-half the total number of strokes. As practice varies somewhat in this respect, it is best to find out in each case, by inquiry, whether the number of strokes made by one plunger or by both plungers in one minute is meant.

In direct-acting steam pumps the piston speed is usually about 100 feet a minute; at least, it is customary to design pumps on this assumption and then to run them faster or slower to suit the required delivery, opening or closing the throttle valve to vary the speed of the pump.

43. Discharge of Pumps.—In formulas for calculating the discharge of steam pumps, the letters employed commonly have the meanings given in the following tabulation:

G = discharge, in gallons per minute;
D = displacement of water cylinder, in cubic inches;
d = diameter of water piston or plunger, in inches;
a = area of water piston or plunger, in square inches;
l = length of stroke, in inches;
n = number of discharge strokes per minute;
S = piston speed, in feet per minute;
E = efficiency of water end.

The discharge of a pump, in gallons per minute, may be found by multiplying together the displacement, the number of discharge strokes per minute and the efficiency of the water end. The product thus obtained is in cubic inches, and must

be divided by 231 to reduce it to gallons. Expressed as a formula,

$$G = \frac{D\,n\,E}{231} \qquad (1)$$

As the displacement is equal to the area of the plunger multiplied by the length of stroke, or $a\,l$, and the area is equal to $.7854\,d^2$, by placing D equal to $.7854\,d^2\,l$, in formula **1** and reducing to its lowest terms, a formula for the discharge in terms of the diameter of the plunger and length of stroke is obtained, as follows:

$$G = \frac{.7854\,d^2\,l\,n\,E}{231}$$

from which $$G = .0034\,d^2\,l\,n\,E \qquad (2)$$

When the piston speed is given, the discharge may be found by multiplying together the area of water piston, the piston speed reduced to inches per minute, and the efficiency. As the product thus obtained is in cubic inches, it must be divided by 231 to reduce it to gallons. Expressed as a formula,

$$G = \frac{12\,a\,S\,E}{231} \qquad (3)$$

Substituting for a its value $.7854\,d^2$ and reducing to its lowest terms, formula **3** becomes

$$G = \frac{12 \times .7854\,d^2\,S\,E}{231}$$

from which $$G = .0408\,d^2\,S\,E \qquad (4)$$

When E is placed equal to unity, that is, when the efficiency of the water end is assumed to be 100 per cent., the foregoing formulas give the theoretical discharge of the pump in gallons per minute. The theoretical discharge is not obtainable in practice.

EXAMPLE 1.—What will be the theoretical discharge, in gallons per minute, of a $14'' \times 10'' \times 20''$ single-acting pump making 80 strokes per minute?

SOLUTION.—The diameter of the steam cylinder, 14 in., is not used in the calculation. As this is a single-acting pump, the number of discharge

strokes is equal to one-half the number of actual strokes, or $n=40$. Further, as the efficiency is assumed to be 100 per cent., $E=1$. The diameter of the plunger is 10 in. and the length of stroke is 20 in.; hence the displacement is $D=.7854\times10^2\times20=1{,}570.8$ cu. in. Finally, substitute the values $D=1{,}570.8$, $n=40$, and $E=1$ in formula **1**, and

$$G=\frac{1{,}570.8\times40\times1}{231}=272 \text{ gal.} \quad \text{Ans.}$$

EXAMPLE 2.—What will be the discharge, in gallons per minute, of a 10″×6″×12″ double-acting pump making 60 strokes per minute, when the efficiency of the water end is taken as 95 per cent.?

SOLUTION.—As the pump is double-acting, the number of discharge strokes is equal to the actual number of strokes, or $n=60$. Substitute $d=6$, $l=12$, $n=60$, and $E=.95$ in formula **2**, and

$$G=.0034\times6^2\times12\times60\times.95=88.13\times.95=83.72 \text{ gal.} \quad \text{Ans.}$$

EXAMPLE 3.—(*a*) What is the piston speed, and (*b*) what is the discharge of a 14″×10″×20″ single-acting pump making 80 strokes a minute, when an allowance of 20 per cent. is made for slip?

SOLUTION.—(*a*) As this is a single-acting pump, the number of discharge strokes is one-half the number of actual strokes, or 40. The piston speed is, then,

$$(40\times20)\div12=66.67 \text{ ft. per min.} \quad \text{Ans.}$$

(*b*) The area of the water cylinder is $a=.7854\times10^2=78.54$ sq. in., and the efficiency of the water end is $100-20=80$ per cent. Substitute $a=78.54$, $S=66.67$, and $E=.8$ in formula **3**, and

$$G=\frac{12\times78.54\times66.67\times.8}{231}=217.6 \text{ gal.} \quad \text{Ans.}$$

EXAMPLE 4.—(*a*) What is the piston speed and (*b*) what is the discharge of a 10″×6″×12″ duplex double-acting pump each side of which makes 60 strokes a minute, the efficiency of the water end being 75 per cent.?

SOLUTION.—(*a*) As this is a double-acting pump, the working strokes equal the actual strokes, or 60. As the length of stroke is 12 in., or 1 ft., the piston speed for each side is $60\times1=60$ ft. per min. As this is a duplex, as well as a double-acting, pump, the piston speed is

$$60\times1\times2=120 \text{ ft. per min.} \quad \text{Ans.}$$

(*b*) Substitute $d=6$, $S=120$, and $E=.75$ in formula **4**, and

$$G=.0408\times6^2\times120\times.75=132.19 \text{ gal.} \quad \text{Ans.}$$

44. Diameter of Plunger.—Formula **4** of the preceding article may be transposed and used to determine the diam-

eter d of a plunger necessary to discharge a given volume of water per minute. When thus transposed, it becomes

$$d=\sqrt{\frac{24.5\,G}{S\,E}}$$

For reasons given in the next article, it is customary, in designing direct-acting steam pumps, to assume a piston speed of, say, 100 feet a minute and, after determining the diameter d of the plunger, to adjust the length of stroke l and the number of strokes n to obtain the best results under the conditions prevailing where the pumps are to be used.

Example 1.—What should be the diameter of the plunger of a pump required to discharge 500 gallons of water per minute when the piston speed is 100 feet a minute and no allowance is made for slip?

Solution.—Substitute the values $G=500$, $S=100$, and $E=1$ in the formula, and

$$d=\sqrt{\frac{24.5\times 500}{100\times 1}}=11.07 \text{ in.} \quad \text{Ans.}$$

Example 2.—What should be the diameter of the plunger of the pump mentioned in example 1, if the efficiency of the water end is assumed to be 80 per cent.?

Solution.—For the sake of the practice it may be assumed that the formula is not available and that the problem must be solved without its use. The volume of water discharged in gallons, per minute, is equal to the product of the area of the plunger in square inches, or 7854 d^2, and the piston speed in inches per minute, or 1,200, divided by 231. In the present example, the plunger must be large enough to discharge $500\div.80=625$ gal. a minute, as the efficiency of the water end is only 80 per cent. Arranging the figures in the form of an equation, $\frac{.7854\,d^2\times 1{,}200}{231}=625$, and transposing, $d^2=\frac{231\times 625}{.7854\times 1{,}200}=153.19$ sq. in. Extracting the square root,

$$d=\sqrt{153.19}=12.38 \text{ in.} \quad \text{Ans.}$$

Example 3.—What should be the diameter of the plunger of a pump to discharge 300 gallons of water a minute when the piston speed is 90 feet per minute and the slip is 25 per cent.?

Solution.—In this case, $G=300$, $S=90$, and $E=1-.25=.75$. Substitute these values in the formula, and

$$d=\sqrt{\frac{24.5\times 300}{90\times .75}}=\sqrt{108.88}=10.4 \text{ in.} \quad \text{Ans.}$$

45. Length of Stroke.—Having determined the proper diameter of plunger for the selected piston speed, it remains to decide on a length of stroke or on the number of strokes per minute in order to obtain the other dimension of the plunger. Thus, if the number of strokes is selected as 40 when the piston speed is 100 feet per minute, the length of stroke will be $100 \div 40 = 2.5$ feet, or 30 inches. Similarly, if the length of stroke is made 24 inches, or 2 feet, the number of strokes will be $100 \div 2 = 50$ per minute when the piston speed is 100 feet a minute.

In practice, the ratio of the diameter of the plunger to the length of stroke varies from 1 : 1 to 1 : 5. Obviously, the greater the ratio the less the number of working strokes in a given time, and the less the number of strokes the fewer times will the valves have to be opened, and the water column stopped and started, and, consequently, the less the jar and wear upon the pump and its connections. For these reasons, a large ratio of length to diameter, with correspondingly few strokes, is preferred for pumps that have to run continuously in hard, rough service, as at mines. If, in example 2, Art. **44**, a ratio of 1 : 2 is selected, the stroke would be $12.38 \times 2 = 24.76$ inches, and the number of strokes would be $(100 \times 12) \div 24.76 = 48.4$ per minute.

46. Approximate Size of Suction and Delivery Pipes. For ordinary work, the velocity of the water flowing through the suction pipe of a pump should not exceed 200 feet a minute, and should not be more than 400 feet a minute in the discharge pipe. However, practice varies in this regard and the maximum allowable velocities in the suction and discharge pipes are sometimes given as 240 and 300 feet per minute, respectively. The diameter of a pipe to carry a given number of gallons of water at a given velocity per minute may be found by the formula

$$d = 4.95\sqrt{\frac{G}{v}} \qquad (1)$$

in which d = diameter of pipe, in inches;
G = gallons of water flowing per minute;
v = velocity of water, in feet per minute.

If the velocity of the water in the suction pipe is taken as 200 feet a minute and that in the discharge pipe as 400 feet a minute, then, by substituting in the formula **1**, and reducing,

$$d_1 = .35\sqrt{G} \qquad (2)$$

$$d_2 = .25\sqrt{G} \qquad (3)$$

in which d_1 is the diameter of the suction pipe and d_2 the diameter of the discharge pipe. The pipes may be made larger than the values found by the formulas, as this will reduce friction; but they should not be made smaller.

EXAMPLE 1.—What should be the diameters of the suction and discharge pipes for velocities of 240 and 300 feet per minute, respectively, when the discharge is 480 gallons per minute?

SOLUTION.—Substitute the values $G=480$ and $v=240$ in formula **1**, and the diameter of the suction pipe is

$$d_1 = 4.95\sqrt{\frac{480}{240}} = 4.95\sqrt{2} = 7 \text{ in.} \quad \text{Ans.}$$

Placing $v=300$, and using the same formula, the diameter of the discharge pipe is

$$d_2 = 4.95\sqrt{\frac{480}{300}} = 4.95\sqrt{1.6} = 6.26 \text{ in.} \quad \text{Ans.}$$

EXAMPLE 2.—What should be the diameter of the suction and discharge pipes of a pump raising 900 gallons of water a minute when the velocity in the pipes named is 200 feet and 400 feet a minute, respectively?

SOLUTION.—Use formulas **2** and **3**, and the respective diameters are

$$d_1 = .35\sqrt{G} = .35\sqrt{900} = 10.5 \text{ in.} \quad \text{Ans.}$$

$$d_2 = .25\sqrt{G} = .25\sqrt{900} = 7.5 \text{ in.} \quad \text{Ans.}$$

These pipes may be made either 10 in. and 7 in., which will increase the velocity and the resistance, or may be made 11 in. and 8 in., which will reduce the velocity and the resistance. The latter course is preferable.

STEAM-END CALCULATIONS

47. Resistances to Pumping.—The resistances to pumping may be classified as follows: (*a*) Static resistance due to the height through which the water must be raised; (*b*) hydraulic resistances caused by the friction of the water in the suction and discharge pipes; (*c*) mechanical resistances caused by the friction of the valves in the pump and pipes

and by the bends and channels in the pump. These resistances may be expressed in pounds per square inch but are commonly expressed in heads, or heights of water column.

With the exception of the mechanical resistance, the other resistances are identical with those of hydraulics, and are calculated in the same way. In hydraulics, the various heads are subtracted from the static head in order to find the available head producing the flow, and, consequently, the quantity of water discharged by a pipe. In pumping, however, the various heads are added to the static head in order to determine the total head to be overcome by the pump. Thus, if the static head is 500 feet and the sum of the other heads is 50 feet, the available head causing the water to flow through the pipe is $500-50=450$ feet; but the total head that must be overcome by the pump under like conditions is $500+50=550$ feet.

48. Static Head.—The static head is equal to the sum of the discharge and suction lifts, and is the vertical distance, in feet, from the level of the water in the sump to the level of the water at the point of discharge. It is the principal resistance in pumping.

EXAMPLE.—The column pipe of a pump is 6 inches in diameter and 1,000 feet long, measured on a slope of 32°. The suction pipe is 8 inches in diameter and 200 feet long, measured on a slope of 5°. (*a*) What is the static head and (*b*) the pressure in pounds per square inch against which the pump must work? What is (*c*) the volume and (*d*) the weight of water in the column pipe?

SOLUTION.—(*a*) The discharge lift is $1{,}000 \times \sin 32° = 1{,}000 \times .52992 = 529.92$ ft. The suction lift is $200 \times \sin 5° = 200 \times .08716 = 17.43$ ft. The static head, which is independent of the diameters of the pipes, is equal to the sum of the discharge and suction lifts, and is

$$529.92+17.43=547.35 \text{ ft. Ans.}$$

(*b*) The corresponding pressure, in pounds per square inch, is

$$547.35 \times .434 = 237.55 \text{ lb. per sq. in. Ans.}$$

(*c*) The diameter of the column pipe is 6 in. or .5 ft. The volume of water in the column pipe is, then,

$$1{,}000 \times (.7854 \times .5^2) = 196.35 \text{ cu. ft. Ans.}$$

(*d*) This volume of water weighs

$$196.35 \times 62.5 = 12{,}271.875 \text{ lb. Ans.}$$

49. Friction Head.—The friction head is the height of a column of water whose pressure is equal to the friction of the water flowing through the pipes. It is the most important of the several hydraulic resistances. The friction head may be calculated by the formula

$$h_f = f\frac{l}{d}\frac{v^2}{2g}$$

in which h_f = friction head, in feet;
f = friction factor;
l = length of pipe, in feet;
d = diameter of pipe, in feet;
v = velocity, in feet per second, of water in pipe;
g = acceleration due to gravity, or 32.16 feet per second per second.

The friction factor may be taken as .02 for smooth pipes and .04 for rough pipes.

Example 1.—What is the friction head in a clean cast-iron column pipe 500 feet long and 6 inches in diameter when the velocity of the water in the pipe is 400 feet a minute?

Solution.—In this case, $d = 6$ in. $= .5$ ft., $l = 500$ ft., and $v = 400 \div 60 = 6.67$ ft. per sec. Insert these values in the formula, and the friction head is found to be

$$h_f = .02 \times \frac{500}{.5} \times \frac{6.67^2}{64.32} = 13.84 \text{ ft.} \quad \text{Ans.}$$

Example 2.—What is the friction head if, in the preceding example, the velocity of the water in the pipe is 300 feet per minute?

Solution.—Here $v = 300 \div 60 = 5$ ft. per sec., whence

$$h_f = .02 \times \frac{500}{.5} \times \frac{5^2}{64.32} = 7.77 \text{ ft.} \quad \text{Ans.}$$

50. When the discharge is given in gallons per minute and the diameter of the discharge pipe is given in inches, the friction head may be found by the formula

$$h_f = f\frac{l\,G^2}{32\,d^5}$$

in which f and l have the same meanings as in Art. **49**, G is the discharge, in gallons per minute, and d the diameter of the pipe, in inches.

EXAMPLE.—A pump discharges 587 gallons of water per minute through a pipe 6 inches in diameter and 500 feet long. (*a*) What is the friction head when the pipe is new and clean? (*b*) What is the friction head when the pipe is old and rough?

SOLUTION.—(*a*) Substitute the given values in the formula, and

$$h_f = .02 \times \frac{500 \times 587^2}{32 \times 6^5} = 13.84 \text{ ft.} \quad \text{Ans.}$$

(*b*) As the friction factor, .04, for rough pipes is twice that for smooth pipes, the friction head for a rough pipe of the dimensions given in the example will be

$$h_f = 2 \times 13.84 = 27.68 \text{ ft.} \quad \text{Ans.}$$

51. Velocity and Entrance Heads.—The velocity head is found from the formula

$$h_v = \frac{v^2}{2g}$$

in which v and g have the meanings given in Art. **49.** At the velocities usual in mining practice, the velocity head is very small—less than .75 foot, or .35 pound per square inch.

The entrance head depends on the shape of the end of the suction pipe. If the end is funnel-shaped, the entrance head is zero; if it is square, this head is equal to one-half the velocity head; and if, as is commonly the case in pumps, the end of the suction pipe is covered with a foot-valve or a strainer, the entrance head may be taken as equal to the velocity head.

52. Curvature Head.—There are always bends in the suction and delivery pipes of mine pumps, and head is required to overcome the resistance they offer to the passage of the water. The curvature head is always very small in mining practice and is usually very much less than the velocity head. It is customary to assume that the curvature head is one-quarter the velocity head when the ratio of the radius R of the bend to the diameter d of the pipe is more than 2 : 1. If the bend is sharp and $R = d$, the curvature head may be assumed as one-half the velocity head. In determining the ratio between them, R and d must be expressed in the same terms, that is, both in feet or both in inches.

53. Mechanical Head.—Strictly speaking, the mechanical resistances should be calculated separately for the suction

valves, discharge valves, and valves in the pipes, and an allowance made for forcing the water through the bends and channels in the pump. Such refinement is uncalled for in mining practice and satisfactory results are obtained if the mechanical head is assumed to be four times the velocity head in the discharge pipe. With this understanding, a reference to Art. **51** shows that the mechanical head is less than 3 feet. The mechanical losses decrease as the number of valve openings in the pump and their areas increase, and are much less at low speeds than at high.

54. A series of calculations on the lines of those explained in the immediately preceding articles will show, in case the vertical lift is 500 feet and the velocity in the discharge pipe (which is rough and 6 inches in diameter) is 400 feet a minute, that the static head is 50 feet, the friction head is 27.68 feet, the velocity and entrance heads are each .69 foot, and the mechanical head is 2.76 feet. The sum of these heads is 531.82 feet. Of this total head, the static head is 500 feet; and all the other heads, or 31.82 feet, comprise only 6.36 per cent. of the static head; and this relation between static head and the sum of the other heads will hold good generally in pumping through shafts in which the vertical lift and the length of the discharge pipe are the same. In other words, when 10 per cent. is added to the static head to allow for friction (sometimes, in problems, an allowance of as much as 25 per cent. is made for friction and "other losses," the nature of which is not specified), the friction as well as the minor heads are amply provided for and calculations are avoided. On the other hand, in pumping through slopes, while the velocity, entrance, and mechanical heads are the same as when pumping through a shaft of the same vertical lift, the friction head is very much greater and cannot safely be covered in a general allowance for "friction and other losses." It must be calculated. Thus, in the pump cited in the opening sentence, if the column pipe is laid in a slope 5,000 feet long with a rise of one in ten, while the static head is 500 feet, as before, the friction head will be 276.8 feet. As the friction head is 57 per cent. of the static head, it is

apparent that an allowance of 10 per cent., or even 25 per cent., of the static head is not sufficient to cover it. For the reason just given it is apparent that in problems involving pumping through slopes, the friction head must be calculated as already stated.

Estimates of the quantity of water a pump will have to handle should be made during the rainy season when the maximum amount of water is entering the mine. Further, an ample allowance should be made for an increase in the volume of water that will have to be handled as the workings expand, pillars are drawn, etc.

55. Theoretical Horsepower.—The theoretical horsepower required for pumping water to a given height may be found by using the formula

$$H=\frac{wh}{33\,000}$$

in which H = theoretical horsepower;
w = weight of water pumped per minute, in pounds;
h = total height water is raised, in feet.

The actual horsepower, of course, is much greater as there is friction to be overcome in addition to the actual work of lifting the water. If the value of h is made equal to the sum of the static head, friction head, curvature head, and mechanical head, the result will approximate the actual horsepower required.

Example 1.—What is the theoretical horsepower required in raising 240 gallons of water per minute through 125 feet?

Solution.—By Art. **35**, the weight of 240 gal. of water is $w=240\times 8.35=2{,}004$ lb. Also, $h=125$ ft. Apply the formula, and

$$H=\frac{2{,}004\times 125}{33{,}000}=7.6 \text{ hp.} \quad \text{Ans.}$$

Example 2.—A pump raises water at the rate of 100 cubic feet per minute. If the total head, including all resistances, is 320 feet, what is the horsepower required?

Solution.—By Art. **35**, the weight of 100 cu. ft. of water is $100\times 62.5=625$ lb. The total head is $h=320$ ft. Apply the formula, and

$$H=\frac{625\times 320}{33{,}000}=6.1 \text{ hp.} \quad \text{Ans.}$$

DUTY OF STEAM PUMPS

56. Definition.—The ratio between the work done by a pump and a certain amount of coal, steam, or heat units used to do the work is called the duty of the pump. During a certain time, say an hour or a day, the pump will raise a quantity of water through a certain height and thus perform a definite amount of work. To do this work, the pump has received from the boilers a certain number of heat units or a number of pounds of steam; or, if the boilers are included as a part of the system, the work has been accomplished by consuming a certain amount of coal. The pump is credited with the work it has performed in the stated time and is charged with the number of heat units, pounds of steam, or pounds of coal it has used in doing the work. It is plain that the economy of the pump or pumping engine is measured by the ratio of the work performed to the steam consumed or the coal burned. Thus, if one pump does 50,000,000 foot-pounds of work with a coal consumption of 100 pounds and another under the same conditions does 36,000,000 foot-pounds and consumes only 60 pounds of coal, the latter is evidently the more economical, since the ratio of work to coal consumption is larger, being $36{,}000{,}000 \div 60 = 600{,}000$ foot-pounds of work per pound of coal as opposed to 500,000 foot-pounds of work per pound of coal.

57. Duty Based on Coal Consumption.—When the duty is based on the consumption, it is customary to assume 100 pounds of coal as the fuel unit; that is, the duty is defined as the number of foot-pounds of work performed for each 100 pounds of coal burned. Then,

$$\text{Duty} = \text{foot-pounds of work} \div \frac{\text{pounds of coal}}{100}$$

or,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 100}{\text{pounds of coal}}$$

Expressed as a formula, the duty is

$$D = \frac{100\, w\, h}{W}$$

in which D = duty;
w = weight of water, in pounds;
W = weight of coal, in pounds;
h = vertical lift, in feet.

The duty based on the coal consumption is of practical value, as it gives an idea of the coal required by a pump of a given type for the performance of a stated quantity of work. It is clear, however, that if a comparison of the merits of two pumps is to be made, the coal must be of the same quality in each case. Further, the boilers supplying steam to the pumps should be of the same type or at least have the same evaporative capacity. This is a point of great importance. One hundred pounds of good bituminous or anthracite coal may, under favorable conditions, evaporate 1,000 to 1,100 pounds of water; that is, furnish that number of pounds of steam to the pump. In many cases, however, the 100 pounds of coal, if of inferior quality and burned under a poor boiler, will not furnish the pump more than 450 to 600 pounds of steam. Under such conditions the duty of the pump based on the coal consumption would not be a fair indication of its efficiency and would not serve as a satisfactory basis for comparing it with other pumps.

Example.—A pump raises 130,000 pounds of water 60 feet and the operation requires the combustion of 25 pounds of coal. What is the duty?

Solution.—Apply the formula, and

$$D=\frac{100\times 130{,}000\times 60}{25}=31{,}200{,}000 \text{ ft.-lb. per 100 lb. of coal.} \quad \text{Ans.}$$

58. Duty Based on Steam Consumption.—In order to avoid the drawbacks incidental to basing the duty of a pump on the coal consumption, it is the custom of some pump makers to specify that the coal consumption shall be estimated on the supposition that a pound of coal evaporates 10 pounds of water, or, in other words, furnishes 10 pounds of steam to the pump. To make this clear, suppose that in a duty trial 32,000 pounds of steam were used by the pump; the duty of the pump would be calculated on the assumption that the coal consumption was $32{,}000 \div 10 = 3{,}200$ pounds, though 5,000

pounds might actually have been used. If 1 pound of coal is assumed to furnish 10 pounds of steam, 100 pounds of coal will furnish 1,000 pounds of steam; hence, the duty based on steam consumption may be defined as the number of foot-pounds of work done by the pump per 1,000 pounds of dry steam supplied it. Then,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 1{,}000}{\text{pounds of steam}}$$

or,

$$D = \frac{1{,}000\, w\, h}{S}$$

in which S is the weight of dry steam supplied in pounds and the other letters have the same meaning as before.

The basis of 1,000 pounds of dry steam is more scientific and better adapted for duty trials than that of 100 pounds of coal, but it is open, nevertheless, to objections. Not only is it difficult to determine the exact weight of dry steam entering the pump, but also 1,000 pounds of steam at 160 pounds pressure will do more work in the cylinder than 1,000 pounds of steam at 60 pounds pressure. If scientific accuracy is sought, the pressure of the steam should be specified in addition to the weight.

EXAMPLE.—A pump lifted 7,920,000 pounds of water 126 feet with 8,100 pounds of steam. What is its duty?

SOLUTION.—Apply the formula, and

$$D = \frac{1{,}000 \times 7{,}920{,}000 \times 126}{8{,}100}$$

$= 123{,}200{,}000$ ft.-lb. of work per 1,000 lb. of dry steam. Ans.

59. Duty Based on Heat Units.—On account of the objections to the basis of comparison then used, a committee of the American Society of Mechanical Engineers in 1891 recommended a new basis for the estimation of duty. Whether the furnace consumes 100 or 200 pounds of coal, whether the steam is at 60 or 160 pounds pressure, wet or dry, the steam cylinders of the pump or pumping engine receive in a given time a definite number of British thermal units. We have seen that if each of two pumps is allowed 100 pounds of coal to do a certain amount of work, one of the pumps may be at a dis-

advantage on account of the poor quality of the coal or the inefficiency of the boiler. If each is allowed 1,000 pounds of dry steam, there may be an inequality because of a difference in the steam pressure in the two cases. If, however, each pump is furnished with an equal number of heat units, each has exactly the same stock in trade, and the merit of each pump can be gauged by the use it makes of the heat units furnished it, that is, by the ratio of the work performed to number of heat units supplied.

60. If a pound of water has a temperature of 212°, it requires 966.1 B. t. u. to change it to steam at atmospheric pressure. If the water has originally a lower temperature or is converted into steam at higher pressure, more B. t. u. are required to accomplish the change. Roughly speaking, if the temperature of the feed and pressure of the steam are not given, about 1,000 to 1,100 B. t. u. are equivalent to a pound of steam. Therefore, 1,000 pounds of steam are equivalent to about $1{,}000 \times 1{,}000 = 1{,}000{,}000$ B. t. u.

Looking at the question in another light, a pound of good coal when burned produces about 13,500 to 14,000 B. t. u. by the combustion. A boiler of fairly good efficiency will utilize perhaps 10,000 of these 13,500 B. t. u., the rest being lost by radiation, in the production of chimney draft, and in other ways. From 100 pounds of coal the boiler is able to extract $100 \times 10{,}000 = 1{,}000{,}000$ B. t. u., which are eventually given up to the pump. It thus appears that 100 pounds of coal and 1,000 pounds of steam are each approximately equivalent to 1,000,000 B. t. u.; for this reason, the committee of The American Society of Mechanical Engineers recommended that the new basis for estimating duty should be 1,000,000 B. t. u.

The heat-unit basis is now very extensively used and is recommended in preference to the others. It may be expressed as follows: The duty of a pumping engine is equal to the total number of foot-pounds of work actually done by the pump divided by the total number of heat units in the steam used by the pump, and this quotient multiplied by 1,000,000. The

heat units in the steam used for driving the auxiliary machinery, such as the air pump and circulating pump of the condenser, if one is used, and the boiler-feed pumps are charged as heat units supplied to the pump.

61. The number of foot-pounds of work done by the pump is to be found as follows: A pressure gauge is attached to the discharge pipe and a vacuum gauge to the suction pipe, both as near the pump as convenient; then the net pressure against which the pump plunger works is equal to the sum or difference in the pressures shown by these two gauges increased by the hydrostatic pressure due to the difference in level of the points in the pipes to which they are attached. In case the gauge in the suction pipe indicates a vacuum, the sum of the pressures indicated by the gauges is taken; but when the water flows into the suction pipe under a head, so that the suction gauge indicates a pressure above the atmospheric pressure, the difference in the two pressures indicated by the gauges is taken.

62. The number of foot-pounds of work done during the trial is equal to the continued product of the net area of the plunger in square inches (making allowance for piston rods), the length of the plunger stroke in feet, the number of plunger strokes made during the trial, and the net pressure in pounds per square inch against which the plungers work.

The pressure corresponding to the vacuum in inches of mercury, indicated by the gauge on the suction pipe, is found by multiplying the gauge reading in inches by .4914, and the pressure corresponding to the difference in the level of the two gauges by multiplying this difference, in feet, by .434. The number of heat units furnished to the pump is the number of British thermal units in the steam from the boilers and is to be determined by an evaporation test of the boilers.

The formula for finding the duty is, then,

$$D = \frac{1{,}000{,}000\,(P \pm p + S)\,A\,L\,N}{H}$$

in which D = duty;
P = pressure, in pounds per square inch, in the discharge pipe;
p = pressure, in pounds per square inch, in the suction pipe, to be added in case of a vacuum and to be subtracted in case of pressure above atmospheric pressure in the suction pipe;
S = pressure, in pounds per square inch, corresponding to difference in level between gauges;
A = average effective area of plunger, in square inches;
L = length of stroke of pump plunger, in feet;
N = total number of delivery strokes;
H = total number of B. t. u. supplied.

EAXMPLE.—A crank-and-flywheel pump has two double-acting water plungers, each 20 inches in diameter and 36 inches stroke. Each plunger has a piston rod 3 inches in diameter extending through one pump-cylinder head.

During a 10-hour duty trial the total heat in the steam supplied to the engine was 35,752,340 B. t. u. and the engine made 9,527 revolutions. If the average pressure indicated by a gauge on the discharge pipe was $95\frac{1}{2}$ pounds, the average vacuum indicated by a gauge on the suction pipe $8\frac{1}{4}$ inches, and the difference in level between the centers of the vacuum and the pressure gauge 8 feet, what was the duty?

SOLUTION.—The area of a plunger 20 in. in diameter is 314.16 sq. in., and the cross-sectional area of a rod 3 in. in diameter is 7.07 sq. in. Since the rod extends through only one end of the pump cylinder, the average effective area of the two ends of each plunger is $314.16-\frac{7.07}{2}=310.63$ sq. in.

The pressure corresponding to a vacuum of $8\frac{1}{4}$ in. is $p=8.25\times.4914=4.05$ lb. per sq. in., and the pressure corresponding to a difference in level of 8 ft. is $S=8\times.434=3.47$ lb. per sq. in.

Since there are two double-acting plungers, the total number of plunger strokes corresponding to 9,527 revolutions is $N=9{,}527\times4=38{,}108$.

Apply the formula, and

$$D=\frac{1{,}000{,}000\times(95.5+4.05+3.47)\times310.63\times3\times38{,}108}{35{,}752{,}340}$$

$=102{,}328{,}800$ ft.-lb. per 1,000,000 B. t. u. Ans.

DISPLACEMENT PUMPS AND AIR LIFTS

DISPLACEMENT PUMPS

63. Definition.—Pumps in which the water is lifted by direct contact between steam or compressed air and the water are sometimes called displacement pumps. This term, however, does not describe the characteristic action of the pump and is purely an arbitrary one, as any piston or plunger pump might equally well be included under the same heading. A number of displacement appliances, known as injectors, ejectors, steam siphons, etc., have been used at one time or another for getting rid of small amounts of water; but their consumption of power, whether compressed air or steam, is so great in comparison with that of ordinary pumps that they are rarely, if ever, used at the present time. Of the larger displacement pumps the only one that ever was of extended application is the pulsometer, but this has been so largely replaced in service by standard sinking pumps that it is seldom seen nowadays.

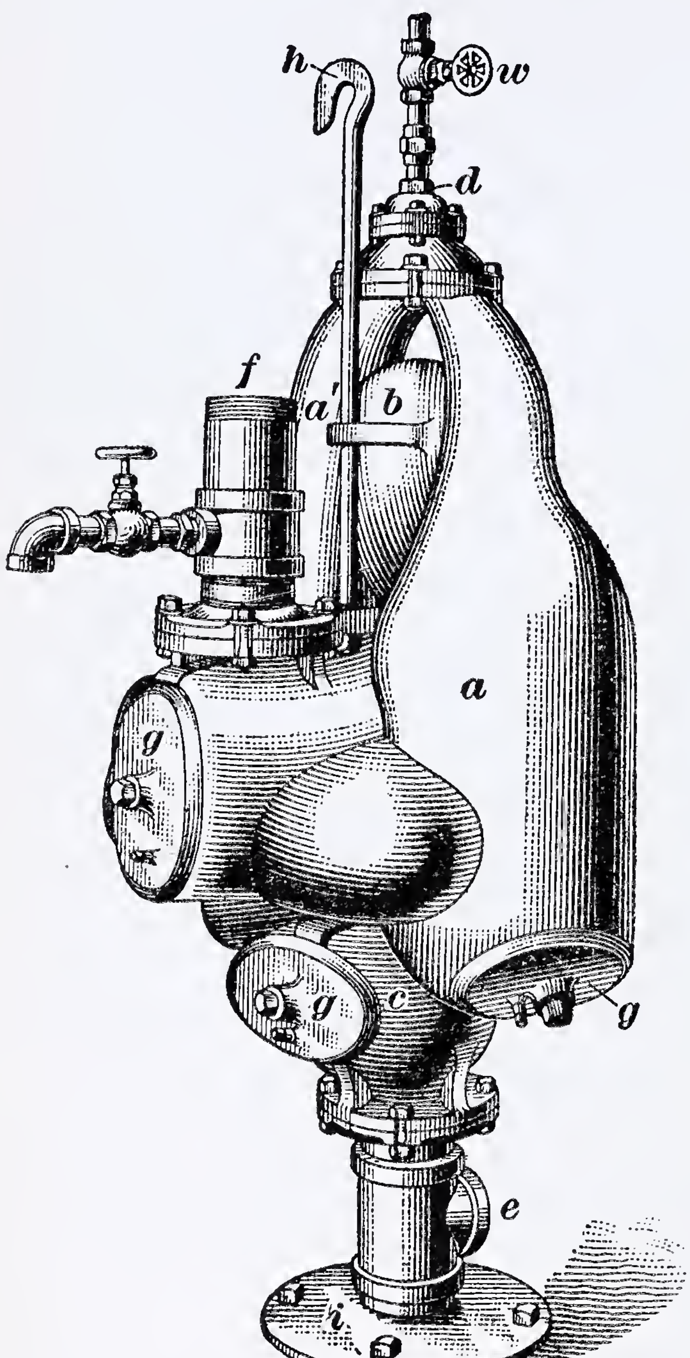


Fig. 7

64. Pulsometer.—In the pulsometer, the water is displaced directly by steam pressure. Fig. 7 is a perspective view of the pulsometer and Fig. 8 is a sectional view. There

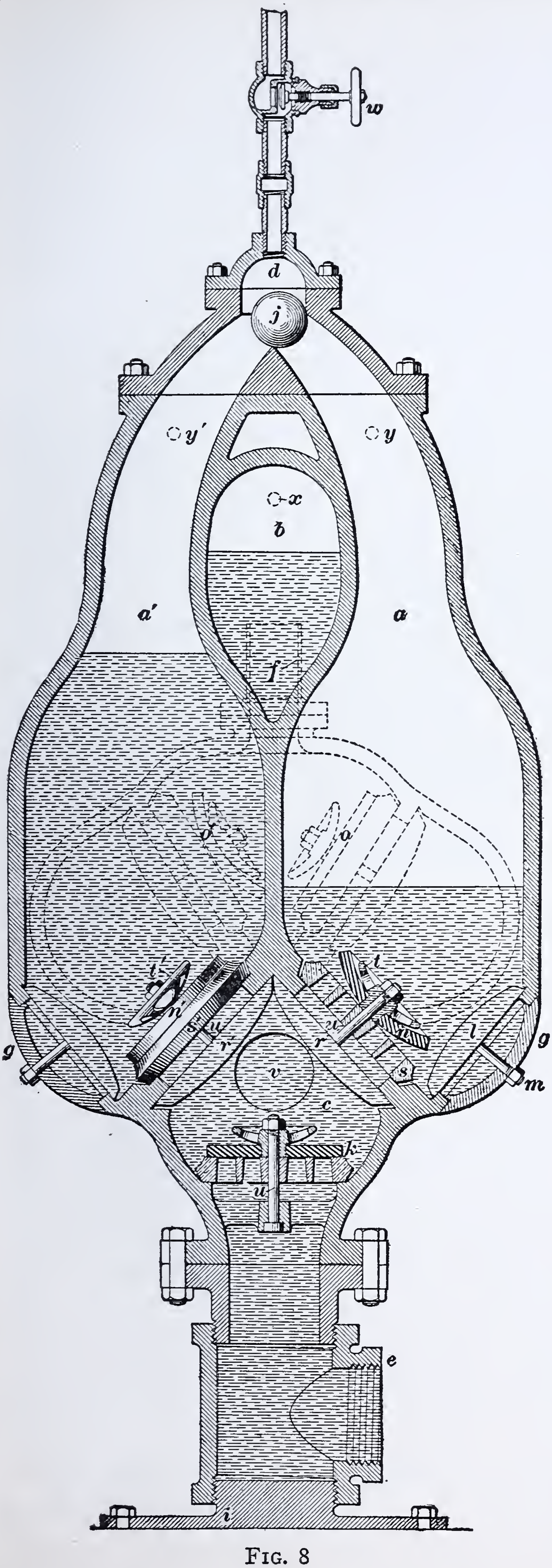


FIG. 8

are two water chambers *a* and *a'* and between them a vacuum chamber *b*, which is not in direct communication with them but is connected with the suction chamber *c*. The steam pipe is attached to the neck piece *d*; the suction pipe is attached at *e*, and the discharge pipe at *f*. There are four oval handholes covered by plates *g*, which are arranged in places suitable for access to the valves and only three of which can be seen in Fig. 7 and two in Fig. 8. When the pump is to be lowered into a shaft, a rope or chain is attached to the hook *h*, but it is always advisable to rest the pump permanently on timbers to assist in sustaining the weight of the pump and the water column and steam-pipe connections; for this reason, the flange *i* is usually attached.

65. The sectional view shown in Fig. 8 is taken so as to pass through the water cham-

bers a, a', the vacuum chamber b, and the suction chamber c. Just below the neck piece d is the ball valve j. The foot-valve k is a flap valve and is just above the tail-pipe connection e; the discharge pipe f is shown by the dotted lines. The handhole plates g are held in place by a yoke l and a bolt and nut m. The suction valves n, n' and the discharge valves o, o' are flap valves, made so that the seats s, s', clamps r, r', and valve guards t, t' are readily removed and replaced. All the flap valves are like those shown in section at k and n. They are held in place by a nut and bolt u. The foot-valve k holds the water in the pump and opens upwards when a new charge is drawn into one of the water chambers a or a'. The suction valves n and n' open whenever water is drawn into the water chambers a and a'. The discharge chamber, shown by the dotted lines in Fig. 8, communicates with the water chambers a and a' and contains the discharge valves o and o'. The suction chamber c forms an indirect communication between the water chambers a and a' and the vacuum chamber b, through the opening v.

66. The action of the pulsometer is as follows: Both water chambers are filled with water to about the height of the water shown at a', Fig. 8. The steam valve w, Fig. 7, is then opened and the steam enters one of the two chambers, as for example a', Fig. 8, the ball valve j then being at the right, as shown. The water in a' will be forced through the discharge valve o' into and up the discharge pipe f and will continue to flow until the water level in a' falls below the edge of the discharge opening. At this point, the steam and water mix in the discharge passage and the steam is condensed, creating a vacuum in a'. The pressure in a being now greater than in a', owing to this vacuum, the ball j is shifted to the left, permitting the steam to enter the chamber a. The water in this chamber is now driven out through o into the discharge pipe f until the steam again mixes with the water in the discharge passage and a vacuum is formed in a, as just described. While this is being done, however, the pressure of the atmosphere the moment the vacuum was formed in a' forced the water

up the suction pipe e, and through the suction valves k and n' into the chamber a', again filling it with water and destroying the vacuum there. When the suction valve of either chamber closes, owing to the shifting of the ball j, the water from the suction pipe enters the vacuum chamber b through the connection v, and is brought gradually to rest by the compression of the air in b, thus preventing a shock owing to the sudden stoppage of the inflowing water. When the water in a has reached the level shown, the steam in a is condensed, the ball j is shifted to the right, and a' becomes the driving chamber.

67. A small air valve x, Fig. 8, admits air to the vacuum chamber b, to replenish that which is lost through leakage and through absorption by the water; two similar valves y and y' admit a small quantity of air to the chambers a and a', respectively, just before the suction begins. This interferes with the suction somewhat, but is necessary in order to regulate and govern the amount of water admitted to the chambers, and to prevent the steam from condensing before the water gets below the edge of the discharge outlet. If there is a partial vacuum formed in a, owing to condensation of the steam, the atmospheric pressure opens the valve y and admits a little air to the chamber. The incoming water compresses this air and soon closes the valve. When the air has been compressed to such an extent as to balance the outside pressure of the atmosphere, the suction valve n will close and no more water can get in. Since the same thing occurs in the other chamber a', it is evident that the amount of air admitted controls the amount of water admitted during the suction period, more water entering when there is less air in the chamber, and vice versa. The admission of the air can be so adjusted that the suction valve in either chamber will close at the instant the ball is shifted to the other side to admit steam.

The cushion of air between the steam and the water prevents them from coming in contact during the forcing process until the water level has sunk below the edge of the discharge

orifice. Air being a poor conductor of heat, the steam does not condense until the mixture of the steam and water has taken place.

68. The pulsometer will raise water by suction to about 75 per cent. of the theoretical height corresponding to the existing barometric pressure. Thus, at sea level, with the barometer standing at 30 inches, it will draw water about 26 feet ($34 \times .75 = 25.5$). It will force water, however, to a height of 100 feet. The pulsometer has no wearing parts whatever except the valves, which are easily and cheaply repaired. It will work in almost any position, and when once started requires no further attention. There are no parts that can get out of order. It will pump anything, including mud, gravel, etc., that will pass through the valves. Its first cost is low and it requires no foundations to set it up. There is no exhaust steam to make trouble and no noise. It uses about twice as much steam as an ordinary direct-acting steam pump of equal capacity.

AIR LIFTS

69. Introduction.—The air lift is a device for lifting water or other fluids by means of compressed air, without the intervention of valves, cylinders, plungers, or other mechanism. It consists of a discharge pipe, or eduction pipe, extending from a point below the surface of the water to be lifted to the point of discharge, and an air pipe that is placed either within or without the discharge pipe and that is led from the main air-supply line to a connection near the lower end of the discharge pipe. Compressed air is forced through the air pipe to the bottom of the discharge pipe and mixes with the water, with the result that the water rises through the discharge pipe and flows out its upper end. The rise of the water in the discharge pipe is due to the energy contained in a volume of compressed air released in the discharge pipe in the form of bubbles; and the driving force causing the lift to operate continuously is due to the difference in weight between the lighter column of

mixed air and water in the discharge pipe and the heavier column of solid water outside it.

70. One of the greatest fields of application for the air lift is the pumping of deep wells, in which service its commercial efficiency is probably higher than that of any other pumping system. It is also used extensively in unwatering mines, pumping water for irrigation, pumping oil and brine wells, and in chemical, metallurgical, and manufacturing processes it has been found of particular value for lifting acids, solutions, oils, and other fluids and semifluids, for which ordinary pumps are not adapted. The air lift is also successfully used for handling mill tailings, and for pumping mixtures of sand and water. The strong features of the air lift are that there are no moving parts, that all the machinery needed is contained in the air compressor which is placed in the engine room, where it operates under the best conditions, and that any number of air lifts can be run from one central air-compressing plant.

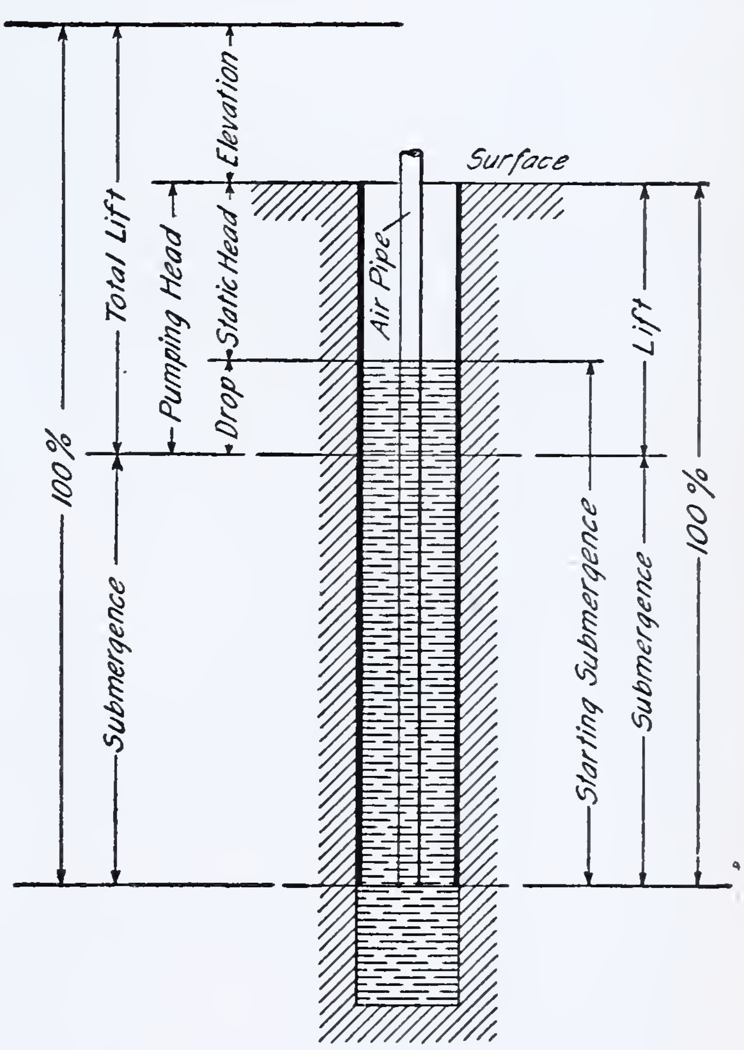


FIG. 9

71. Terms Used in Air-Lift Work.—The various terms used in air-lift pumping are shown graphically in Fig. 9. *Static head* is the distance from the ground surface or top of the well casing to the surface of the water in the well when no pumping is being done. *Drop* is the distance the water level in the well is lowered when pumping begins. *Pumping head* is the distance from the ground surface or top of the well casing to the surface of the water in the well when pumping is going on. It is equal to the sum of the static head and the drop. *Elevation* is the distance above the ground surface or top of the well casing to which the water must be lifted.

Total lift is the distance the water is raised from the level during pumping to the point of discharge, and is equal to the sum of the elevation and the pumping head. If discharge is at the ground surface, it is equal to the pumping head. *Submergence* is the distance from the surface of the water, during pumping, to the point at which the air picks up the water. *Starting submergence* is the distance from the surface

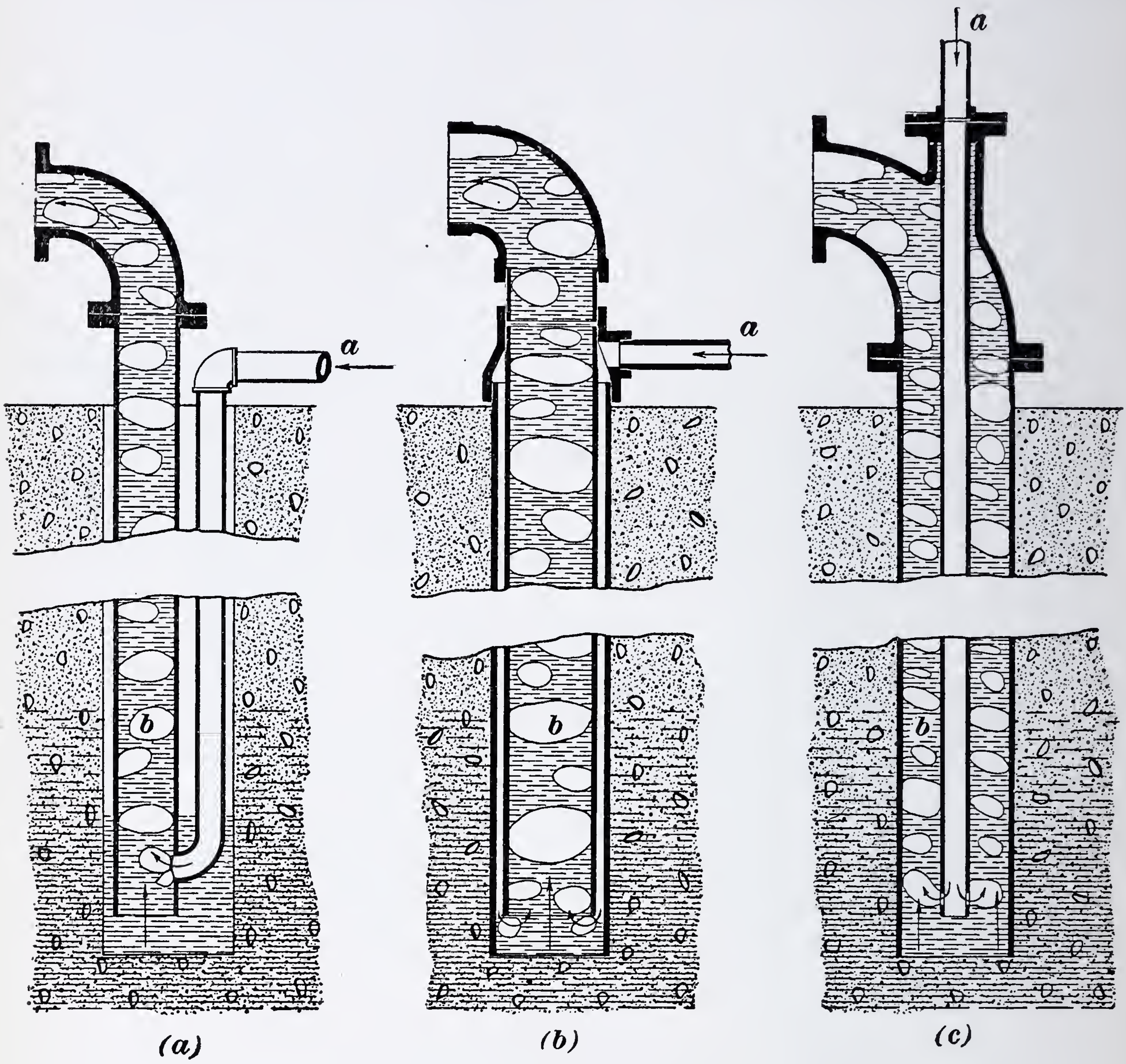


Fig. 10

of the water, before pumping begins, to the point where the air picks up the water. It is equal to the sum of the drop and the submergence. *100 per cent.* is the vertical distance that the air travels with the water, from the point of introduction to the discharge level. If there is some elevation, it is the sum of the submergence and the total lift. If discharge takes place at the ground surface, it is the sum of the submergence and the lift.

72. Straight Air Lift.—The straight air lift, often called the Pohle air lift, and the three systems of connecting the air pipe *a* and eduction pipe *b* are shown in Fig. 10 (*a*), (*b*), and (*c*). In the outside system, shown in (*a*), the air is carried down outside the eduction pipe, into which it is discharged a short distance above the bottom. In the reservoir system, shown in (*b*), an eduction pipe is suspended in the well casing, allowing the air to pass down between the two and mix with the water at the bottom of the eduction pipe. In the central pipe system, shown in (*c*), the air is carried down through a pipe suspended inside the eduction pipe and allowed to discharge into the water through an open end. In all of these systems the principle is much the same. Pressure is built up in the air passage until it is sufficient to overcome the head due to submergence, when a large bubble of air passes into the eduction pipe. This flow of air from the pipe temporarily reduces the air pressure, so that the weight of water in the well outside of the eduction pipe, due to submergence, shuts the air off and a plug of water follows the plug of air up into the eduction pipe, until the compressor has had time to build up the air pressure and the air again breaks through. This process repeats itself and becomes rhythmic in its action, forming in the eduction pipe a succession of air bubbles and water plugs that are alternately discharged at the surface. Although at times a more or less continuous discharge is obtained, it is considered a phenomenon of the system, and the cause is not thoroughly understood.

73. The theory of the air lift and reason for submergence are that by mixing the air and water in the eduction pipe the column is made lighter than a column of solid water; therefore, the pressure of the column of air and water in the bottom of the eduction pipe is less per square inch than that of the solid water outside in the well, and an upward flow is thus created in the eduction pipe. The bubbles of compressed air in their ascent prevent any slipping back of water; for, as the air progresses upwards to the point of discharge, it expands on its way, and in proportion as the overlying weight

of water decreases, the volume of the air increases, the pressure of the air becoming less, until, at the surface, it reaches that of the atmosphere. The air lift should be a perfect expansion pump, but natural laws prevent this in ordinary straight air installations. Some of the reasons for low efficiency in straight air lifts are as follows:

1. If the difference in pressure between the air and the water is small, where the water enters the eduction pipe, the air will flow into the water at low velocity. In the open-end air lift, such as shown in Fig. 10, the air must travel some distance toward the surface in the eduction pipe before it expands sufficiently to form a plug. This results in a loss in submergence.

2. As the bubble of air travels toward the surface and the pressure above it decreases, it must expand. As it is confined in the pipe, it can do this only in a vertical direction, which results in increased velocity and friction and a greater displacement in the eduction pipe.

3. The flow at the point of contact between the gas or liquid and the walls of the pipe is retarded, and as water is heavier than air, it is retarded the most, and is pulled back around the air bubble. The bubble becomes elongated and at times slips through, joining the preceding one and leaving the water behind, resulting in slippage and consequent loss of efficiency.

4. Air, like any other confined gas, is always seeking a chance to escape, and any inequalities in the flow passage or abrupt changes in direction encourage this slipping effect.

74. Sullivan Air-Lift System.—Experiments and careful investigation of different methods of mixing air and water, proper proportioning of piping, and other refinements of the previously described principles of the straight air lift have resulted in obtaining considerable improvement in efficiencies. In the apparatus developed by the Sullivan Machinery Company, provision is made to secure a complete mixture of air and water, forming an emulsion at the point where the air is injected into the water, thereby permitting each small particle of air to start its lifting effect at once; also, a Venturi tube,

or throat, is provided just above the mixer to increase the velocity and give a jet effect at this point, and a more nearly constant upward pressure on the ascending column. The eduction pipe is also designed so as to provide, as far as possible, for proper expansion of air, and to prevent excessive velocities as the point of discharge is approached. The use of a tank that acts as a separator, or booster, and a device called a compound mixing tube, or relift pump, for elevated discharge are some of the features that greatly improve the efficiencies of operation of this system of handling water.

75. The Sullivan standard air-lift foot-piece, illustrated in Fig. 11, is constructed of bronze, with an air passage *a* leading from the outside and connected to a perforated tube *b* below a Venturi throat *c*, so that the air is broken into small jets, thereby producing an emulsion of air and water and increasing the velocity in the throat of the tube. By having a large number of holes in the mixing tube *b*, a mixture containing more or less air, depending on the lift, is made automatically without any change in the pump. An opening *d* below the mixing tube permits the dislodging and escape of any sand or dirt lodging in the tube when pumping is stopped, by the back flow or waste of the falling column of water.

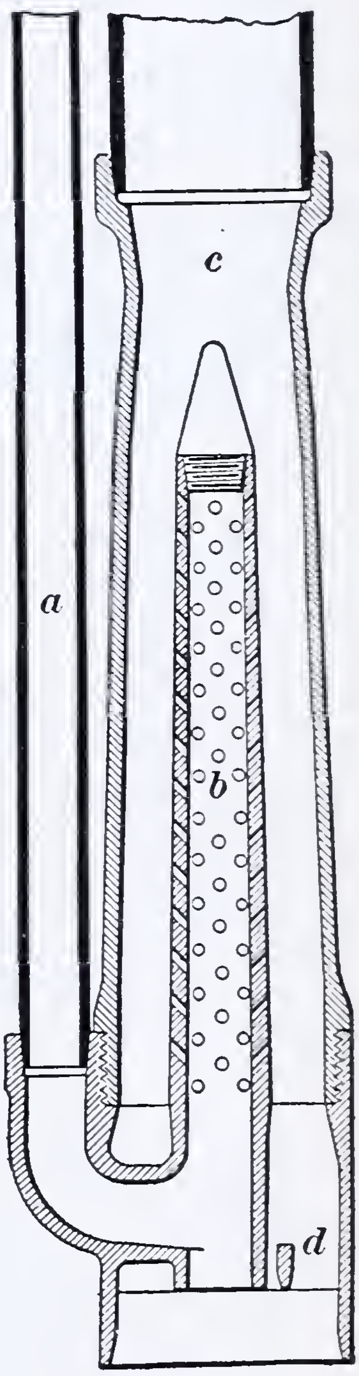


Fig. 11

The standard well top illustrated in Fig. 12 embraces flanges *a* and *b* to seal the top of the casing *c*, a water discharge nipple *d*, an umbrella separator *e*, and a jam nut *f*. There is an air-line nipple *g* with stuffing-box *h*, the air line being provided with a tee *i* having a place for a pressure-gauge connection, and an adjustable air cock *j*.

76. When water is to be discharged at an elevation above the surface, either near to or at a distance from the well, considerable gain in efficiency is secured by the use of a separa-

tor or booster, which consists of an air-tight shell or tank that receives the combined air and water discharged at the surface. The discharge enters the tank at the top, on one side (that is, at a tangent to the periphery), at a considerable velocity and assumes a downward swirling effect that results in a perfect separation of the air from the water. The water leaves the separator through an outlet at the bottom, also at a tangent to the periphery, while the air passes to the top of the separator. The air may be disposed of in one of three different ways: (*a*) It may be allowed to escape to the atmosphere through the vent valve. (*b*) It may be returned to the intake of the compressor; and as its temperature is lower than that of the surrounding atmosphere, some gain in volumetric efficiency results. (*c*) When the water is discharged at an elevation above the surface, a considerable gain results by discharging the air from the separator through a mixing tube in the base of the riser pipe.

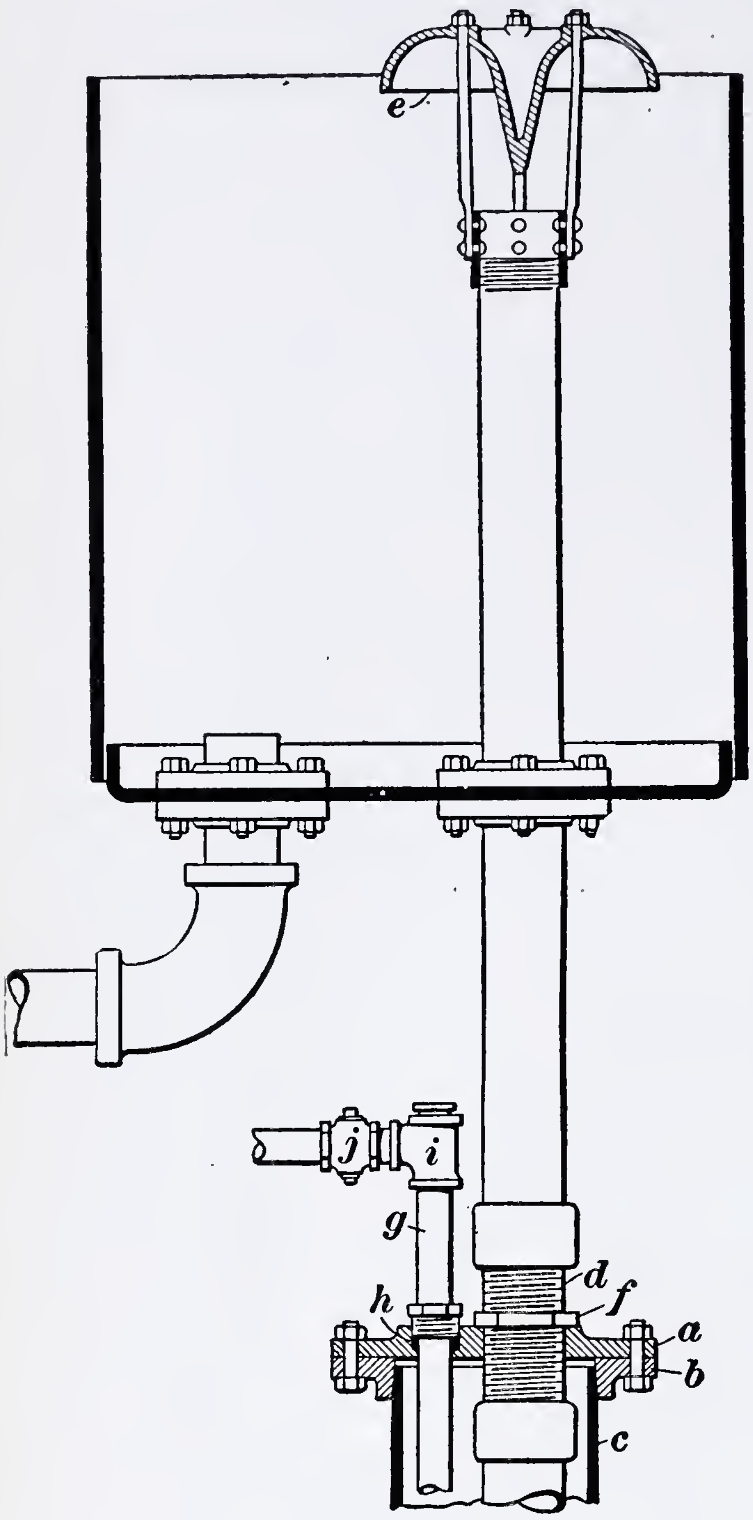


FIG. 12

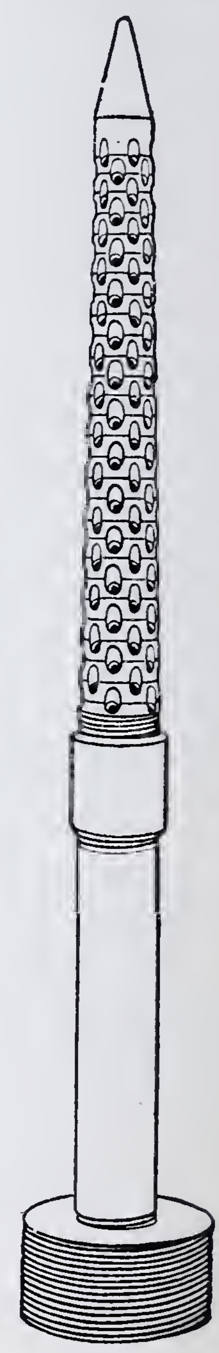


FIG. 13

77. Mixing Tube.—As illustrated in Fig. 13, the mixing tube is a device for injecting air into the water column at some point in its passage between the surface and an elevated storage tank. Its purpose is to lighten the discharge column of water, thus reducing the head against which the air lift must operate. The mixing

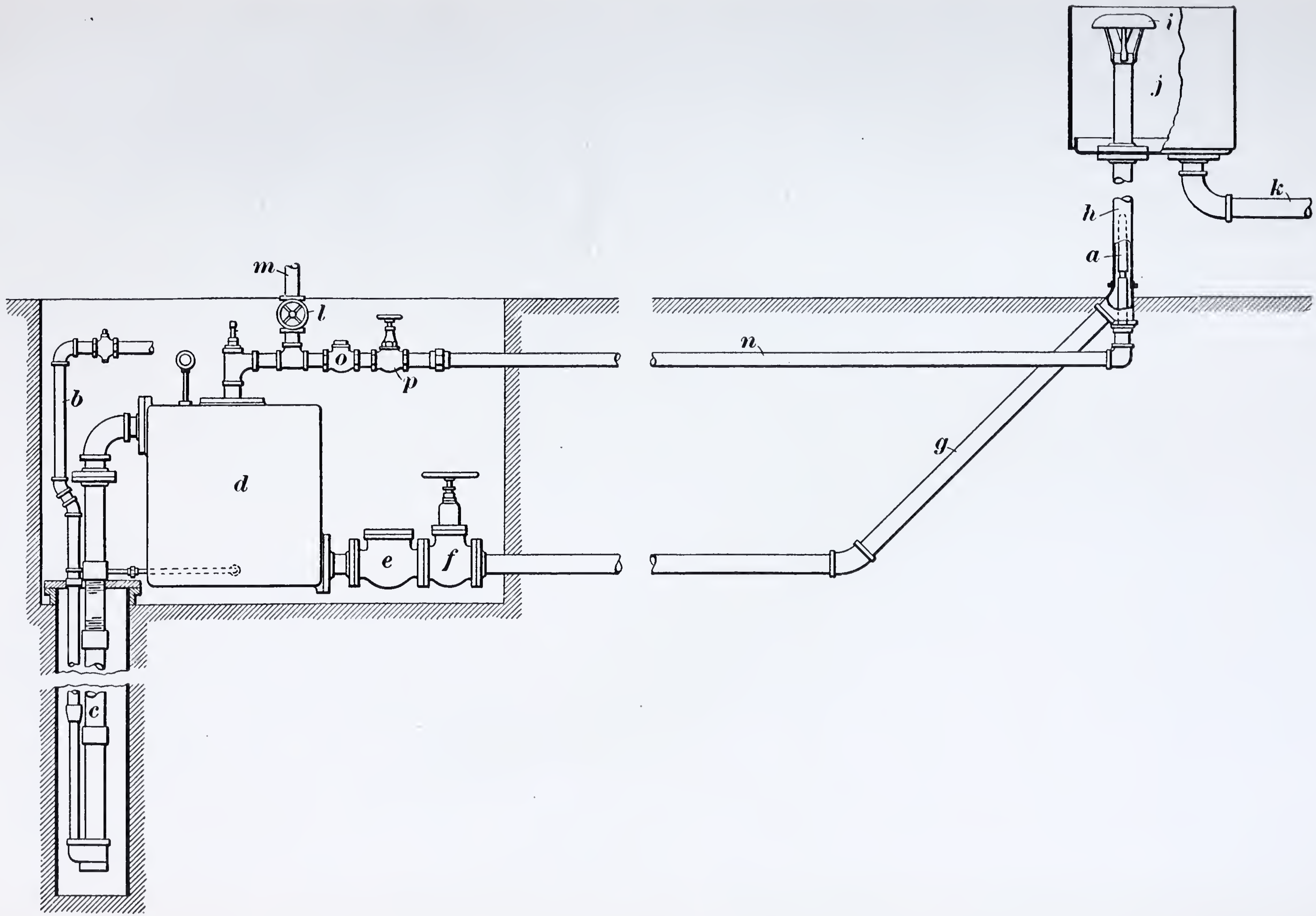


Fig. 14

tube is made of bronze, well filled with perforations, and the base of the tube is provided with a plug to fit the **Y** of the riser pipe. The position of this mixing tube in the system is shown at *a*, Fig. 14.

78. Typical Air-Lift System.—The details of the Sullivan air-lift pumping system, embracing the numerous devices previously described, are shown in Fig. 14. It is considerably more elaborate than the straight air lift, and the resulting efficiency is correspondingly greater. The air from the compressor is led to the pump in the well through the pipe *b* and the discharge from the eduction pipe *c* enters the booster, or separator, *d*. The water from the separator passes out through the swing check-valve *e* and the gate valve *f* into the discharge pipe *g* and at the point *a* is mixed with air from the mixing tube, whence it is carried through the riser *h* to the umbrella *i* and falls into the tank *j*. It then flows to the reservoir through the pipe *k*. The air from the separator *d* may be allowed to escape to the atmosphere through the valve *l* and the pipe *m*; or, it may be sent to the mixing tube *a* by way of the pipe *n* containing the check-valve *o* and the globe valve *p*. The pipes *g* and *n* are arranged so as to drain back into the separator *d*.

79. As the lift increases, the necessary ratio of submergence to lift decreases. Thus, for a lift of 50 to 100 feet, the best percentage of submergence is 65 to 70; for a lift of 150 feet, the best percentage of submergence is 60 to 65; for a lift of 200 feet, the submergence should be 55 to 60 per cent.; and for lifts of from 400 to 700 feet, the submergence should be 40 to 45 per cent. For a given lift and quantity of water, the size of the discharge pipe increases as the submergence increases. The diameter of the air pipe depends on the quantity of air required and its pressure, but should be of such size as to permit a velocity of 20 to 25 feet a second for the air. It is not possible to devise a formula of general application to all conditions; yet if the static head, elevation, size of well, and air pressure are known, manufacturers of air lifts will, because of their experience, be able to devise a system that will work satisfactorily.

POWER AND CENTRIFUGAL PUMPS

Serial 3030 Edition 1

POWER PUMPS

TYPES AND APPLICATION

1. Drive for Power Pumps.—As explained in another Section, a power pump is a reciprocating pump whose water piston or plunger is driven by belt and pulley, gearing, chain, or connecting rod from some source of power which may be a steam, gas, or gasoline engine, or an electric motor.

The great majority of power pumps are driven by electric motors; in fact, when the term electric pump is used, an electrically driven power pump is usually understood to be meant. In order to reduce the speed of the electric motor to that suitable for the pump piston, the armature shaft carries a small gear that meshes with a larger gear on the shaft that actuates the connecting rod that drives the pump, as is shown in many of the following illustrations. The great advantage of electric drive is that the pump may be placed at any distance from the generator. An independent power line may be strung from the generator to the motor of the pump, or power may be taken from a circuit supplying current to other machines. Thus, in mining practice, it is customary to take the power for the pump from the trolley line on the haulage roads. In any event, whether power is drawn from an independent or a common line, the cost of conveying electric power is much less than that of conveying steam power, and the operating cost is also much less. While it is true that the simplest method of driving a power pump is by belt and pulley,

yet there are many instances where there is not sufficient room for this arrangement. In all such cases, a geared electrically driven pump economizes space and, consequently, saves in the cost of housing, installation, etc. All motor-driven power pumps may be arranged for automatic control, which is of particular advantage in case the pump is in service where the water supply is intermittent, as it enables the pumpman to attend to numerous pumps without danger that any one of them will run dry, or that it will remain idle when there is water to be pumped.

2. Where steam power only is available, an appreciable saving in the cost of operation, as compared with a direct-acting steam pump, can be obtained by using a steam engine for driving a power pump. A direct-acting steam pump consumes from 100 to 150 pounds of steam per horsepower per hour, while a fairly efficient simple engine consumes but 30 to 40 pounds. Therefore, a steam-driven power pump will handle a given amount of water with from one-third to one-half the steam required by a direct-acting pump. This saving in steam reduces the consumption of water and coal and the amount of labor otherwise required to operate the pumps, and also reduces the cost of the equipment, as a smaller plant can be used. Steam power pumps are often belt-connected to the engine.

The gas or gasoline engine is a convenient source of power for driving isolated pumps where electricity or steam is not available. The engine may be belt-connected to the pump, or the power may be transmitted by gearing.

A great advantage possessed by all power pumps is that the speed of the engine or motor need not be fixed, as the proper speed for the pump may be obtained by using gears or pulleys of suitable diameters.

3. Belt-Driven Piston Power Pump.—A single-cylinder, double-acting, belt-driven, piston, power pump is shown in Fig. 1. This style of pump is usually constructed in small sizes, the capacities not exceeding 200 gallons per minute against heads of less than 200 feet. It is particularly adapted

to handling dirty or gritty water, such as prevails in mines, because all the valves are quickly accessible from the outside by loosening the nuts *a* and removing the yokes *b* that hold the plate covers in place. This construction permits the quick removal of particles of coal or other materials that occasionally lodge on the valve seats. The main frame *c* is of the heavy box type of construction, and contains bored cross-head guides and babbitt-lined shaft bearings. The pump cylinder *d*, which contains all the valves on seats within its body, is

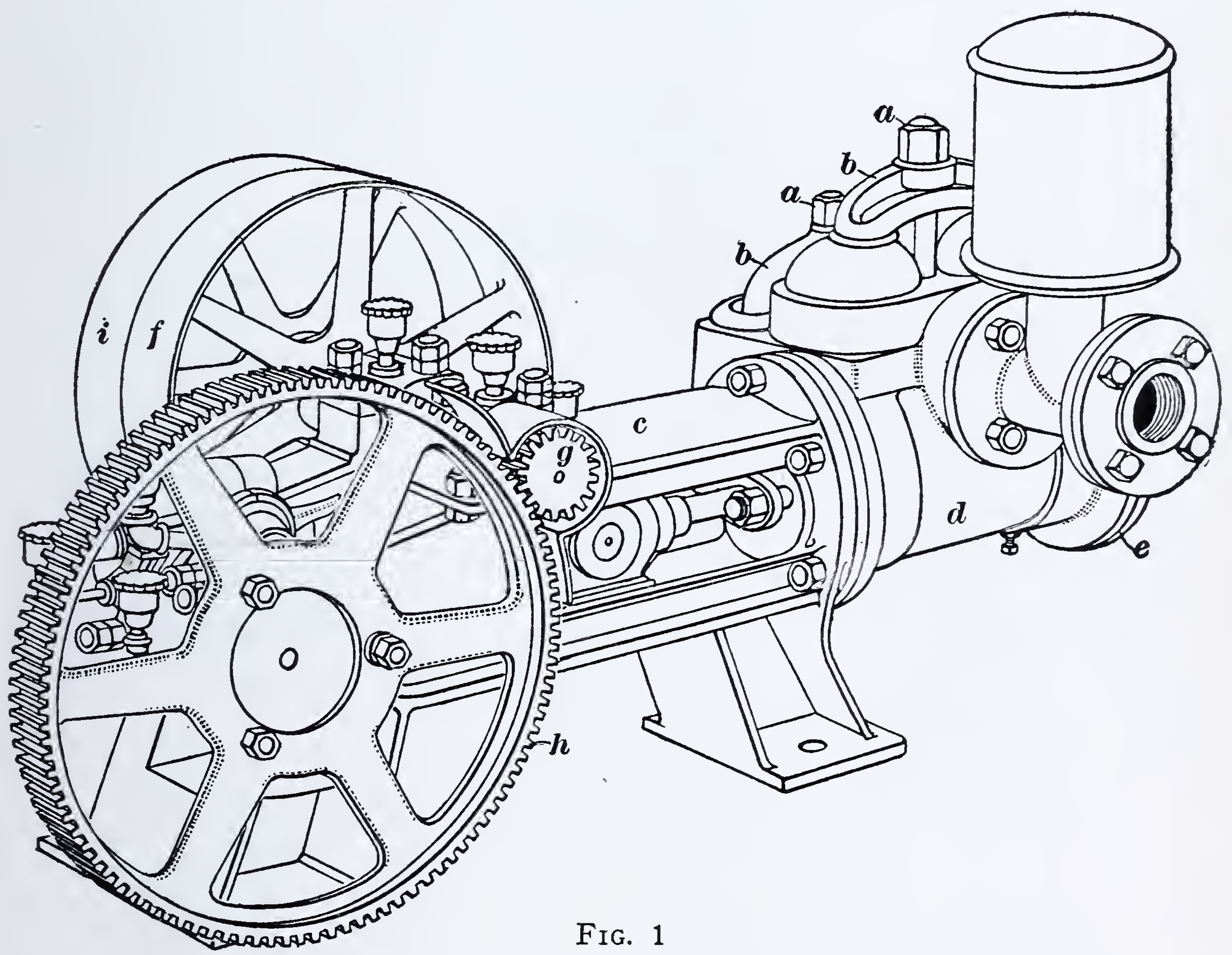


FIG. 1

usually made of cast iron, and may be fitted with a bronze or acid-resisting liner. The piston and piston rod likewise may be constructed of bronze or acid-resisting metal. The piston is packed with fibrous packing and is easily accessible by removing the cylinder head *e*. The construction of the pump is such that the entire cylinder may be unbolted and removed without disturbing the foundation or interfering in any manner with the driving mechanism.

The pump is driven by a belt through the pulley *f*, which is fixed to the same shaft as the pinion *g*. This pinion meshes

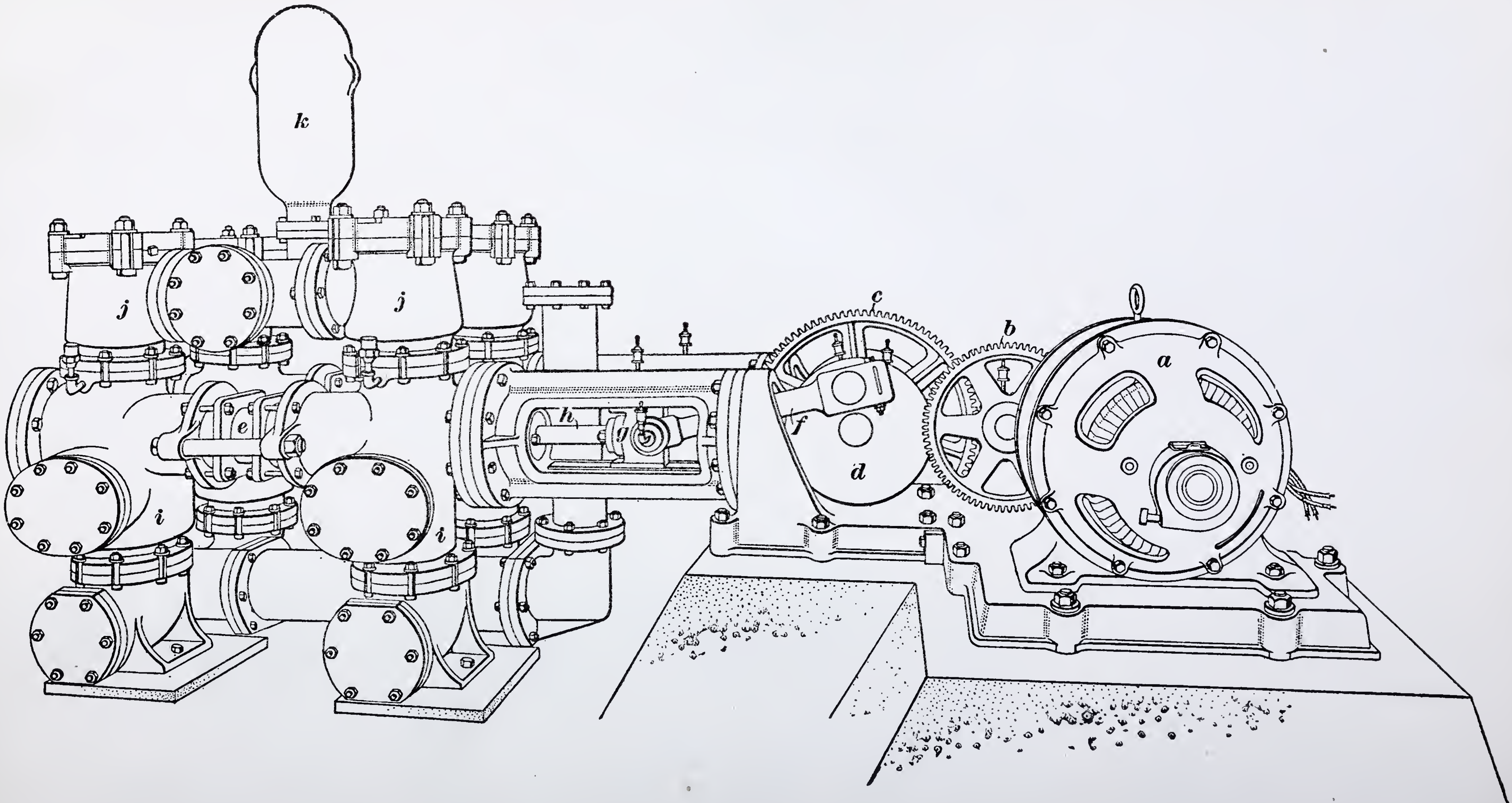


FIG. 2

with the main gear *h*, which is fixed to the crank shaft. The object of the gearing is to reduce the rotative speed of the crank shaft below that of the motor, or pulley, shaft. In some pumps, a double set of reduction gears is used instead of the single set shown at *g* and *h*. The pump is stopped by shifting the belt from the driving pulley *f* to the idler, or loose, pulley *i*, or by stopping the motor that drives the belt. The moving parts, such as gears and crank, should be fitted with guards to prevent bolts and the like from falling into them and damaging them, and to eliminate, as far as possible, the possibility of accident to the pump-runner. All bearings are lubricated by means of grease cups, although the crank-shaft bearings may be lubricated by the splash system.

4. Center-Packed Plunger Power Pump.—A duplex, double-acting, center-packed, plunger power pump, driven through double reduction gearing by an electric motor, is shown in Fig. 2. Pumps of this style are intended for large capacities and high heads, and as they are designed for mine service, they are made extremely rugged and with all parts very accessible. The pump can be taken apart in small sections, thus permitting of easy transportation down shafts and along narrow passageways underground. A substantial foundation is required for the maintenance of an accurate alinement of the moving parts, which is necessary for the successful operation of the pump. When properly installed, this pump is efficient and satisfactory in units having capacities of more than 500 gallons per minute, and for heads of 400 feet or more.

In the pump shown in Fig. 2, the electric motor *a* drives the gear *b* by means of a small pinion on the end of the armature shaft. A second pinion, fixed to the same shaft as the gear *b*, meshes with the gear *c*, which in turn is fixed to the same shaft as the crank disk *d*. Thus, the high rotative speed of the motor is reduced to a suitable speed of the crank disk *d*, and the pump plunger *e* is given a reciprocating motion through the agency of the connecting rod *f*, crosshead *g* and piston rod *h*. The suction valves are located in chambers *i* forming the lower

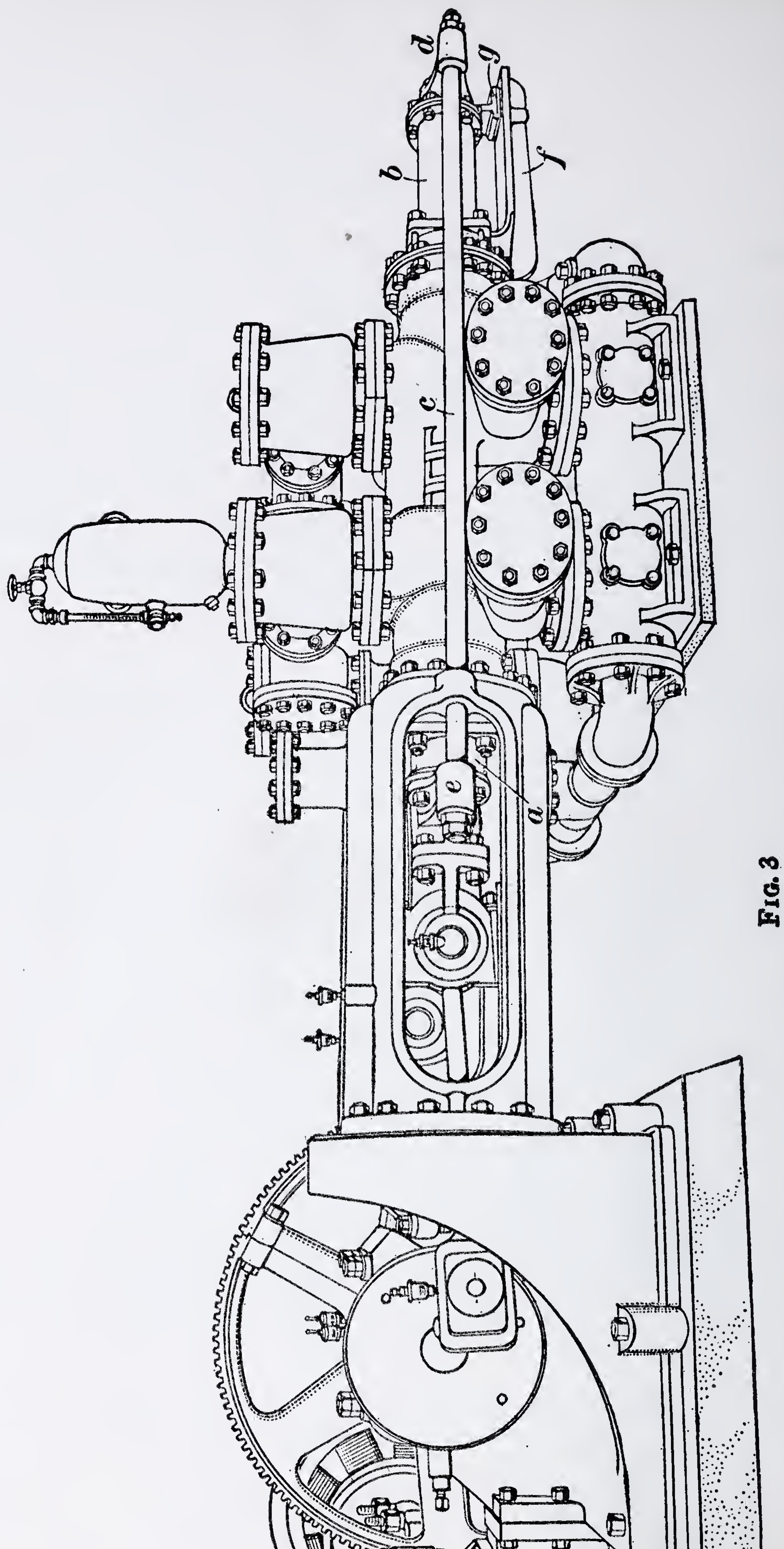


FIG. 3

parts of the working barrels and the discharge valves are in detachable chambers *j* at the top. The air chamber *k* is fitted to the discharge pipe to reduce the shocks that result from the pulsating movement of the water on the discharge side of the pump.

5. Outside-Packed Plunger Power Pump.—An outside-packed plunger power pump having a duplex water end is shown in Fig. 3. The several features of its design are very much like those of the pump just described, except that the plungers *a* and *b* are joined by outside rods *d* and *e*. A tail guide *f* forms a bearing for the shoe *g* and supports the plunger *b*. A single set of reduction gears is used between the motor *h* and the pump. Both the power and the water ends are built in sections so that, in event of accident to any part, replacement may be made more cheaply than is possible with pumps in which a number of parts are combined in one casting. The outside-packed plunger is recognized as the standard design for handling to best advantage the various kinds of bad water found in mines.

6. Vertical Single-Acting Power Pumps.—Vertical power pumps consist of a series of pump cylinders placed side by side drawing water from a common suction pipe and discharging it into a common column pipe. The pump plungers are actuated by connecting rods from a common crank shaft as will be seen later. While pumps of this type may have but one or two cylinders, they usually have three, four, or five. When there are three cylinders, the driving cranks are set 120 degrees apart; when there are four cylinders, 90 degrees apart; and when there are five cylinders, 72 degrees apart. In each of these arrangements, the discharge strokes of the plungers overlap one another with the result that a smooth, uniform flow of water is obtained with high efficiency and absence of water hammer. The source of power for driving the pump may be of any of those already mentioned.

A single-acting triplex power pump is shown in Fig. 4, (*a*) being a side view, partly in section, and (*b*) a cross-section on the center line of one of the pump cylinders. A belt on the

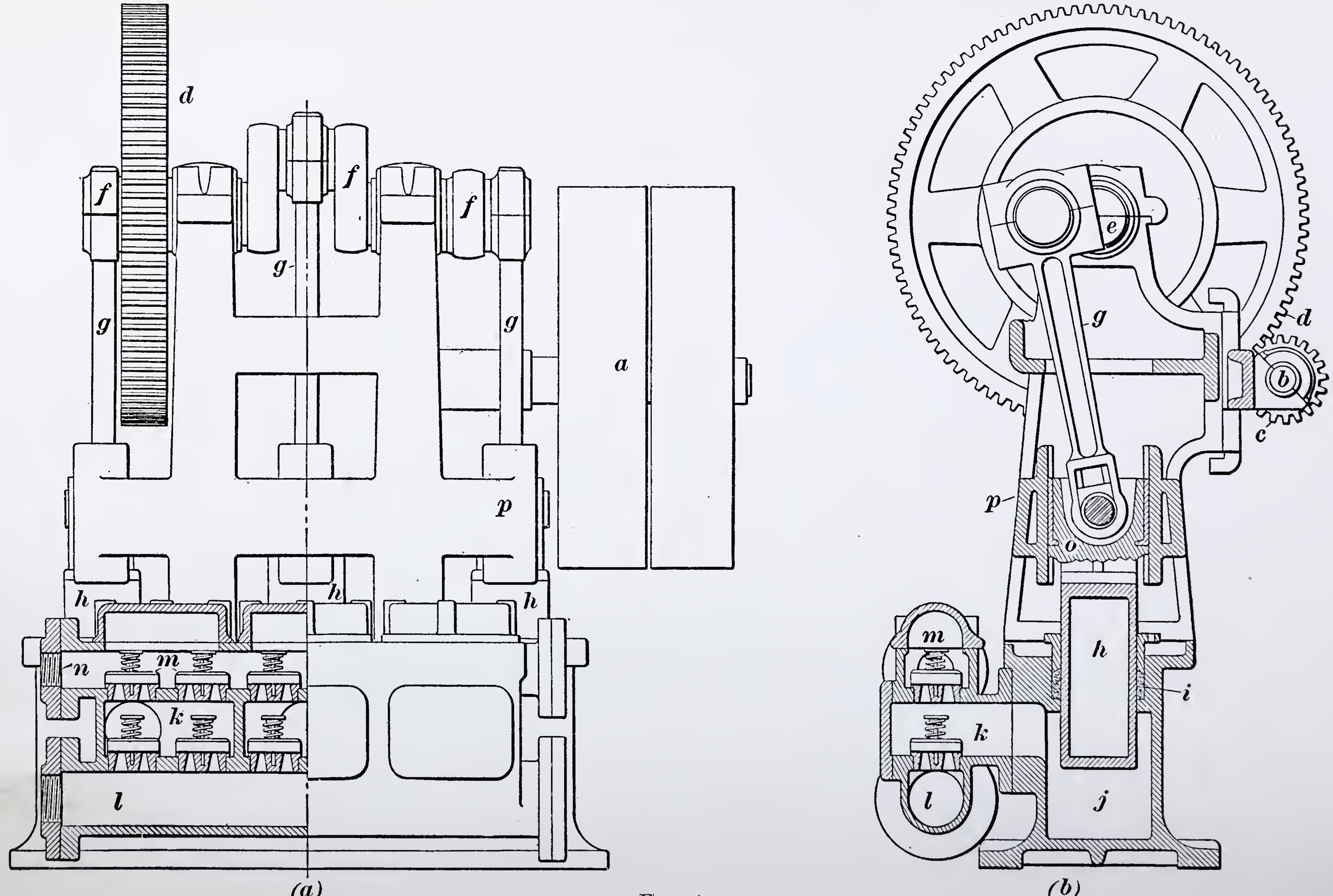


Fig. 4

tight pulley *a* drives the shaft *b* which carries a pinion *c* that meshes with the large gear *d* on the crank-shaft *e*. On the crank-shaft, are three cranks *f* that give motion to the connecting-rods *g* and thus to the pump plungers *h*. Each plunger works through a stuffingbox *i* in the top of the pump barrel *j*. On the upward stroke of the plunger, water is drawn into the pump barrel through the valves *k* from the suction pipe *l*. On the downward stroke of the plunger, the valves *k* close and the water is forced out through the valves *m* into the discharge pipe *n*. The crosshead *o* forms a connection between the plunger and the connecting-rod and slides in guides *p* in the columns that support the crank-shaft bearings. If the pump room is high enough to hold it, this type of pump may be used in mines, provided the head is not excessive and the quantity of water to be handled is not more than about 200 gallons per minute.

7. The triplex single-acting pump shown in Fig. 5 is driven through a single set of reduction gears by an electric motor *a* mounted on the same subbase as the pump. The suction-valve chambers *b* and the discharge-valve chambers *c* are of the pot type, and the valves are accessible when the caps *d* are removed. These valves are placed on opposite sides of the pump barrel *e*, and as a result the flow of water into the pump barrel and out of it is always in one direction, there being no reversal of flow as in the pump described in the preceding article. Because of this feature, the pump may have a relatively high speed with a correspondingly low slip. The water end of the pump is made in sections for ease in handling in narrow places and to facilitate and cheapen the making of repairs. Pumps of this kind are usually limited in capacity only by the cost as compared with centrifugal pumps of large capacity. They are most satisfactory for mine work and, when properly installed, will continue in service for many years with a minimum cost for upkeep.

8. High-Head Power Pump of Large Capacity.—A typical example of a power pump of large capacity for use in deep mines is shown in Fig. 6. It is built for handling 1,000

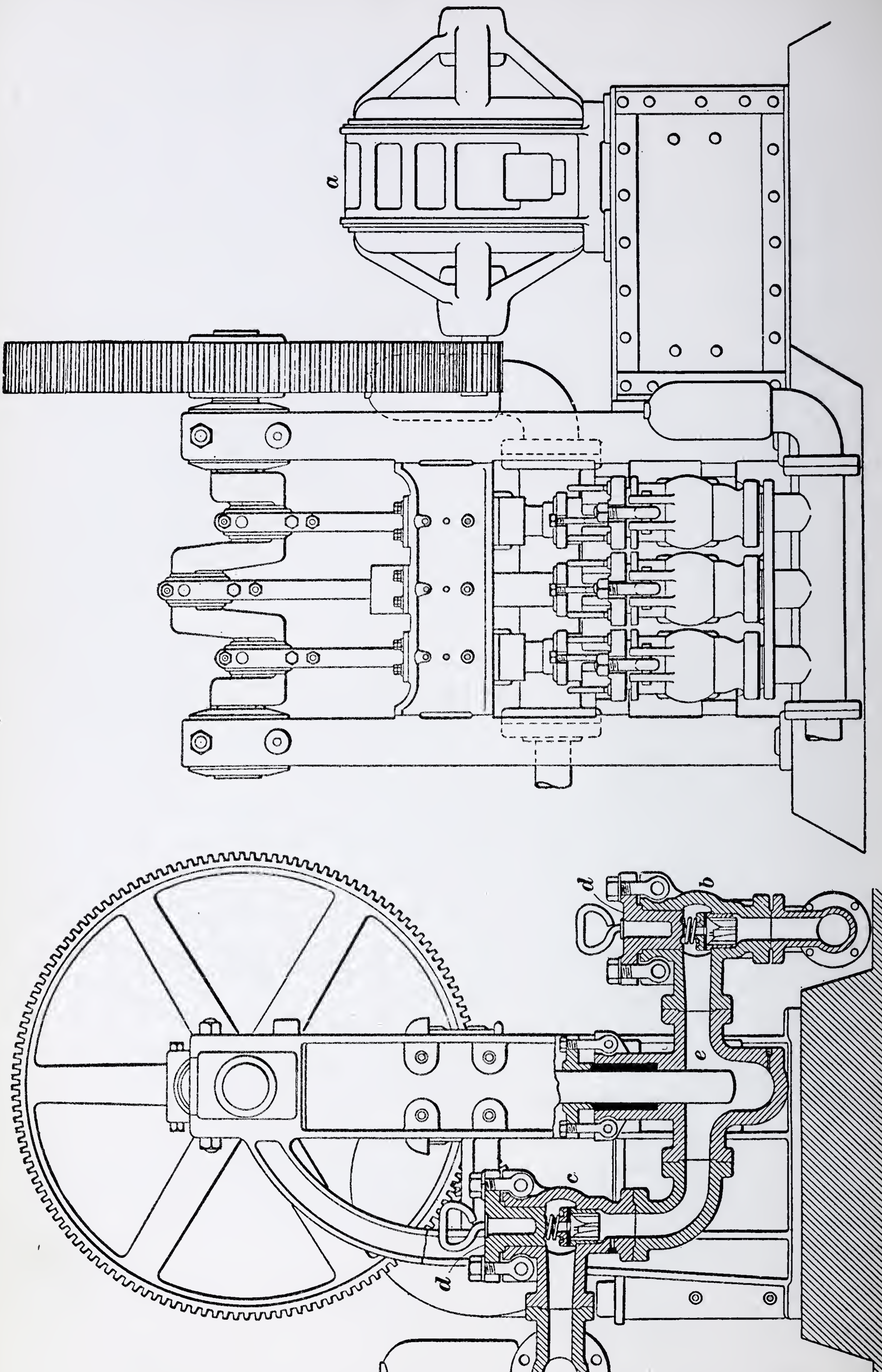


FIG. 5

gallons of water per minute against a head of 1,000 feet. The water fittings are of bronze and the water passages are lined with cement, so that acid water may be pumped without injury to the pump. The pump is belt driven, and the reduction gears are of the herringbone pattern, as shown at *a*. This form of gear-tooth not only gives greater strength but also reduces the noise when the gears are in motion. The suction pipe *b* is fitted with an air chamber *c* and the suction valves are contained in the chambers *d*. The discharge valves

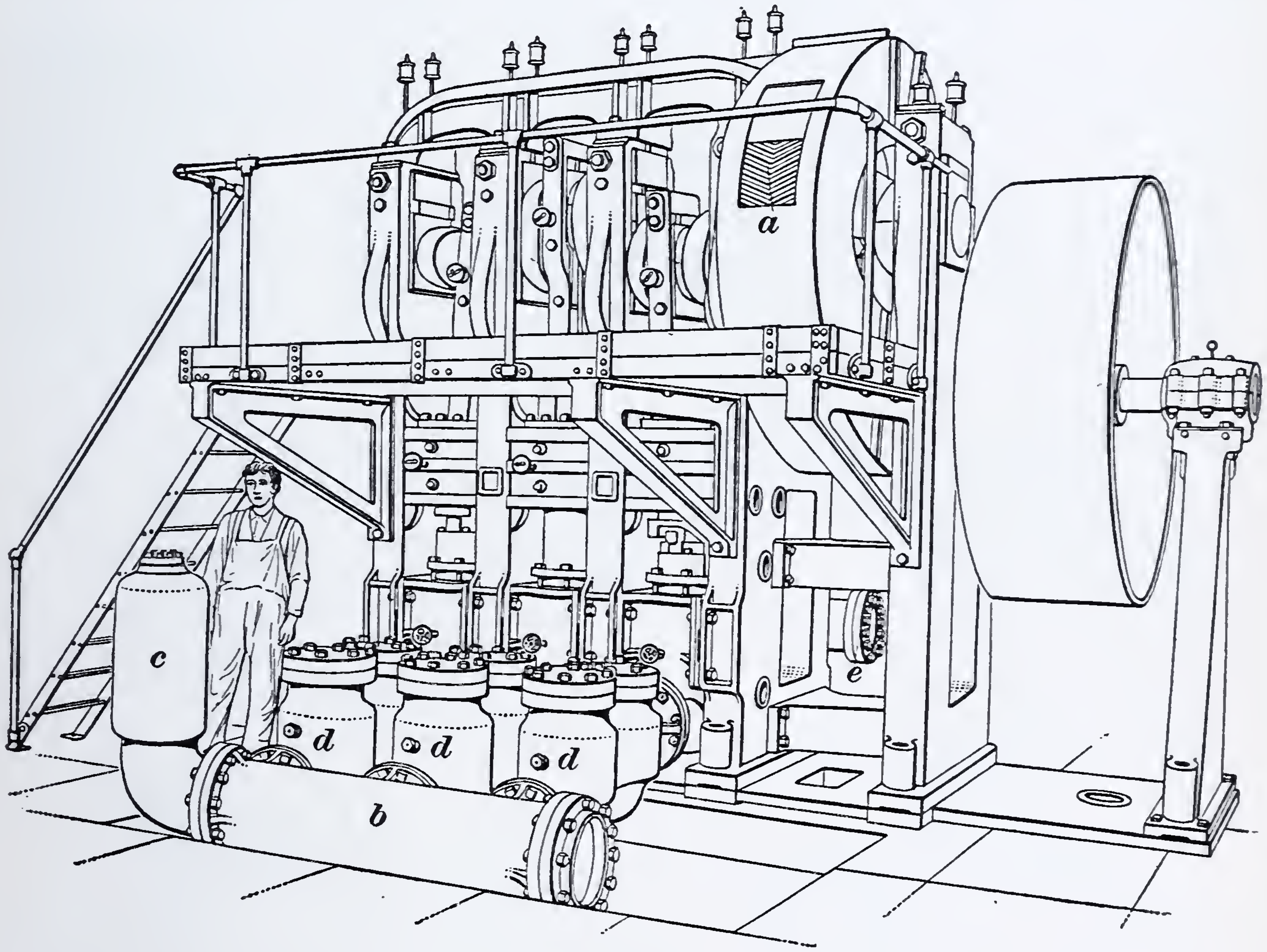


Fig. 6

and discharge pipe *e* are at the farther side of the pump. The entire design of the frame, base, and driving mechanism is heavy and rigid, as might be expected of a pump for this class of service.

9. Horizontal Single-Acting Power Pumps.—One of the strong features of the horizontal power pump is that the suction valves can be placed in such a position as to keep the plunger at all times submerged; thus, the pump barrels are always primed and are sure to be filled at each stroke. As in vertical pumps, both power and water ends are sectionalized

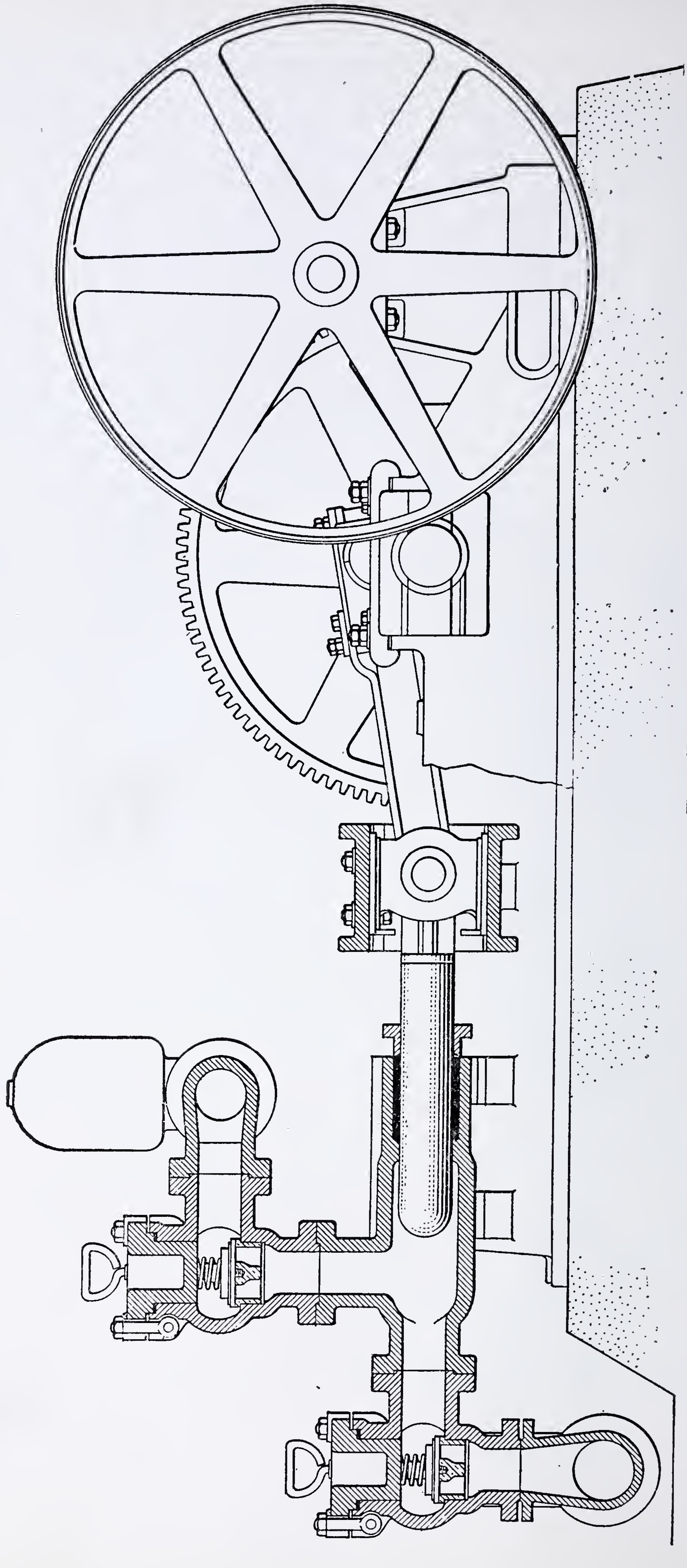


Fig. 7

to facilitate handling in shafts and narrow places and the making of repairs. A horizontal single-acting power pump arranged for belt drive is shown in Fig. 7. The pot-chamber design indicates that the pump is adapted to working against

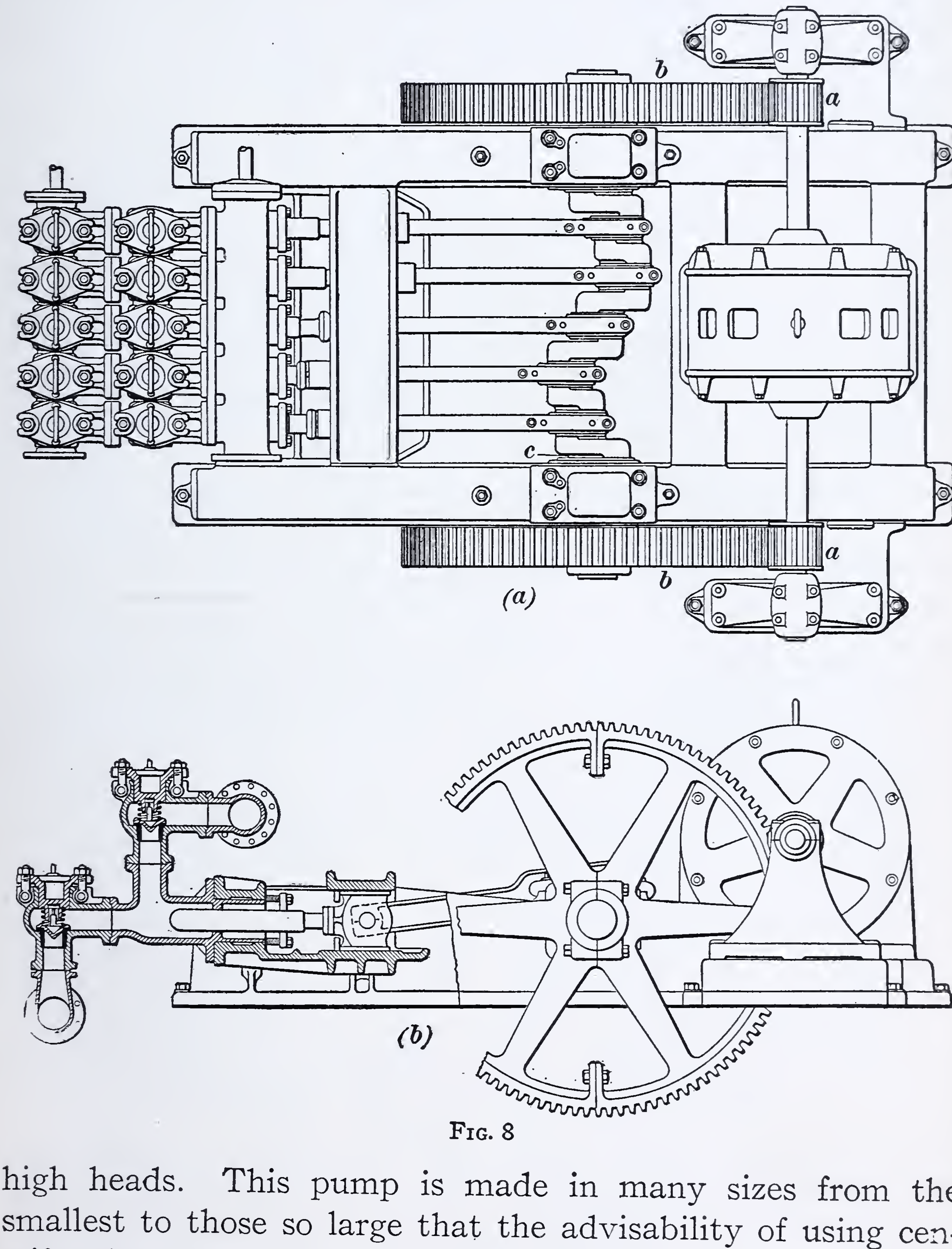


Fig. 8

high heads. This pump is made in many sizes from the smallest to those so large that the advisability of using centrifugal pumps in their place has to be considered.

10. Where the service is exceptionally severe and the pump has several working barrels, the strains often become too

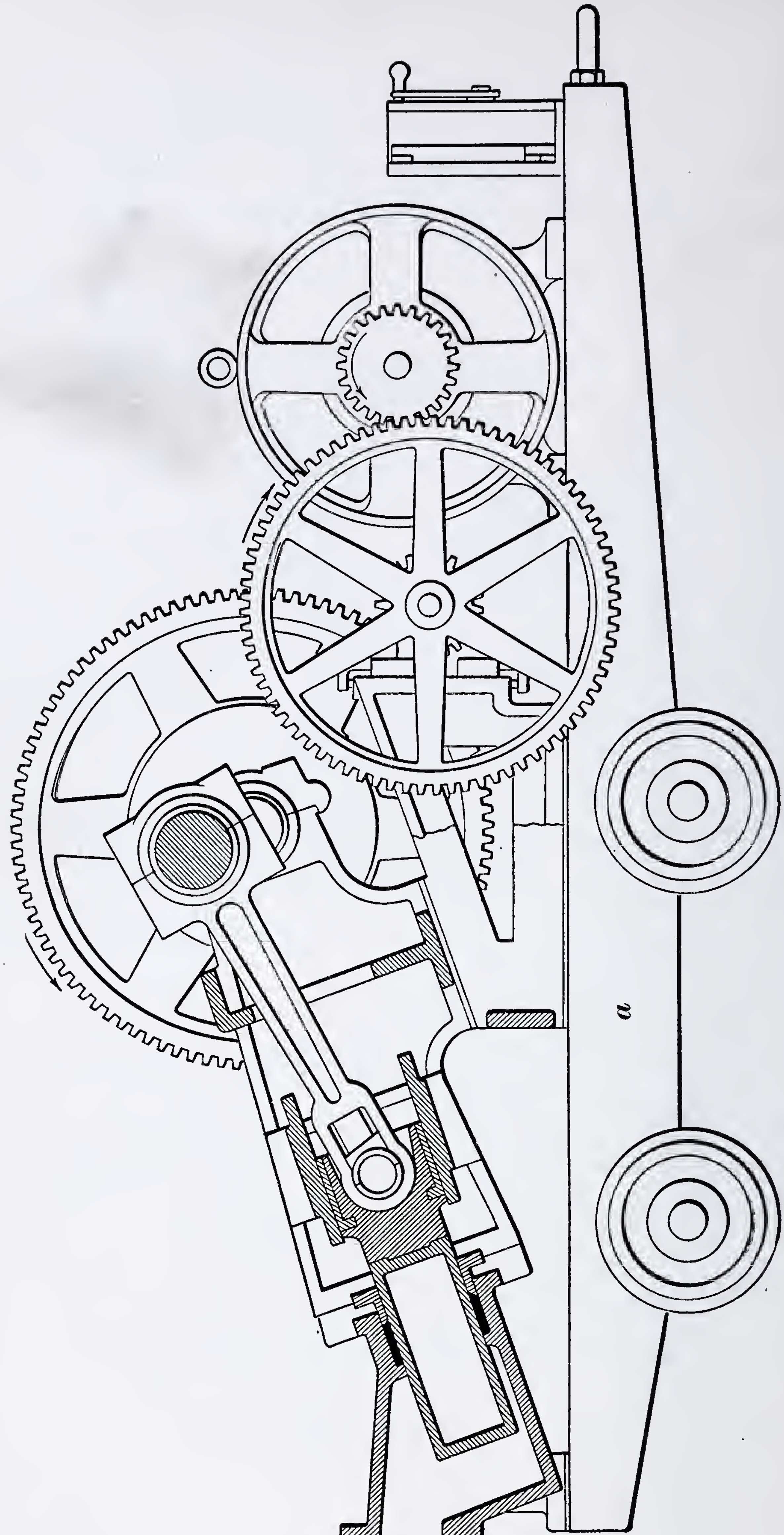


FIG. 9

great for safety when the power is transmitted through a single pinion. In such a case the arrangement shown in Fig. 8 (*a*) and (*b*) may be used. Two pinions *a* are fixed to the ends of

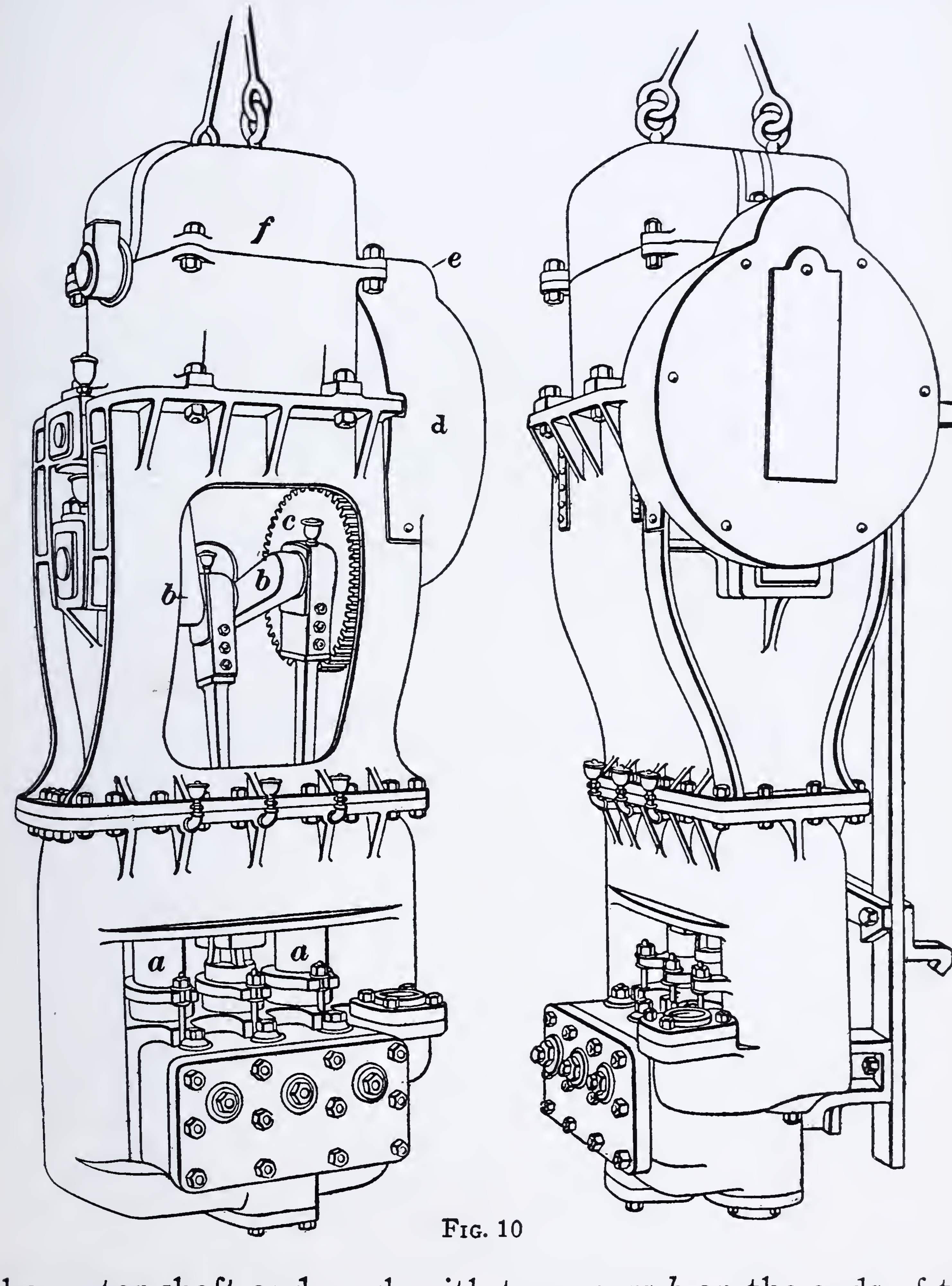


FIG. 10

the motor shaft and mesh with two gears *b* on the ends of the crank-shaft *c* of the pump, thus distributing the pressures over a greater area of metal.

11. Portable Sump Pump.—One of the most convenient forms of mining pump for handling the small quantities of

water that collect in local dips is the portable sump pump shown partly in section in Fig. 9. It is mounted on a truck *a* that can readily be transported from one place to another. The truck is made of steel **I**-beams and channels in order to secure the permanent alinement of the pump and its motor. The truck may be propelled by the pump motor; or it may be a plain truck provided with a drawbar for attachment to an electric locomotive. As the pressures in this kind of service are not high, heavy rubber hose of large size is often used for both suction and discharge pipes.

12. Sinking Pump.—A triplex electric sinking pump is shown in Fig. 10. The outside-packed plungers *a* are driven by cranks *b* that are rotated through the gears *c*, *d*, *e*, by the incased motor *f*. The suction pipe of sinking pumps is made of wire-wound rubber pipe. The connection between the discharge pipe and the pump is made so that it may be quickly disconnected, if necessary to hoist the pump out of the way at shot-firing time.

POWER PUMP INSTALLATION

13. Foundations.—A suitable foundation is necessary for the satisfactory operation of a motor-driven power pump. The foundations may be made of concrete or brick and should be heavy and rigid enough to support the pump and motor bedplate at all points, and maintain this support permanently. Pump bedplates will not and should not be expected to maintain proper alinement of pump and motor if they are not thoroughly supported. The pump and motor base should be placed upon the foundation with suitable leveling wedges under each end and at the middle of the pump and motor bedplate. The bedplates or bases should then be leveled each way, by use of the wedges to raise or lower them as may be necessary, until the desired level position is obtained. If the motor is connected to the pinion shaft by a flanged coupling, the bolts should not be placed until the edges of the faces of the couplings are in line, and exactly agree, and the pump and the motor operate freely, both separately and when coup-

led together. If the pump and the motor are connected by double reduction gears, the pump is not lined up properly until the motor pinion and the motor gear mesh together with even pressure throughout the length of the tooth, and until the teeth mesh to such an extent as to allow one to three thicknesses of tissue paper to pass between the teeth without pinching. The bedplates should then be grouted in place so that they will be absolutely rigid, and so that all parts of the pump and motor base will be thoroughly supported.

14. Power Transmission.—As already stated, power pumps may be operated by steam, compressed air, and gas or gasoline engines, or electric motors, prime movers that are described in other Sections. When chain or gear drive is used between the motor and the pump, the size and detailed dimensions are usually fixed by the pump builders and constitute an integral feature of the pump design. When belt drive is used, it is frequently necessary for the pump operator to arrange his own power transmission. This is better done in consultation with the pump manufacturer who, if told the size of the driving and driven pulleys, the distance between the centers of these pulleys, the speed of the motor pulley, type of pump, horsepower to be transmitted, etc., will designate a belt of the proper width and thickness for the work in hand.

15. Gears and Pinions.—Where it is desired to reduce noise or vibration, rawhide pinions may be used to advantage. Their use is limited, however, to the transmission of about 110 horsepower. They are especially suited to high-speed transmission, but to get satisfactory service from them they should be protected from moisture and oil, which is usually a difficult matter to accomplish in mines. Pinions made of compressed fiber paper also give good results, as far as the elimination of noise and vibration are concerned, and they have the advantage of being stronger than rawhide and not being affected by moisture and oil. Where paper or rawhide pinions are employed it is necessary to use a larger pitch and a wider face than would be necessary for transmitting a given amount of power by a steel or manganese-bronze pinion; but

it is advisable to use the same size of steel or manganese-bronze pinion as of rawhide or paper, so that they can be interchanged if necessary. The herringbone gear and pinion, shown in Fig. 11, are excellent for high speed service. These gears operate with less noise and transmit more power for the same width of face than the common spur gear. A steel gear and a manganese-bronze pinion make a good combination for high-speed and large power transmission. Metal gears, except when engaged with rawhide pinions, should be oiled to prevent possible cutting. Flaked graphite only should be used as a lubricant for rawhide pinions.

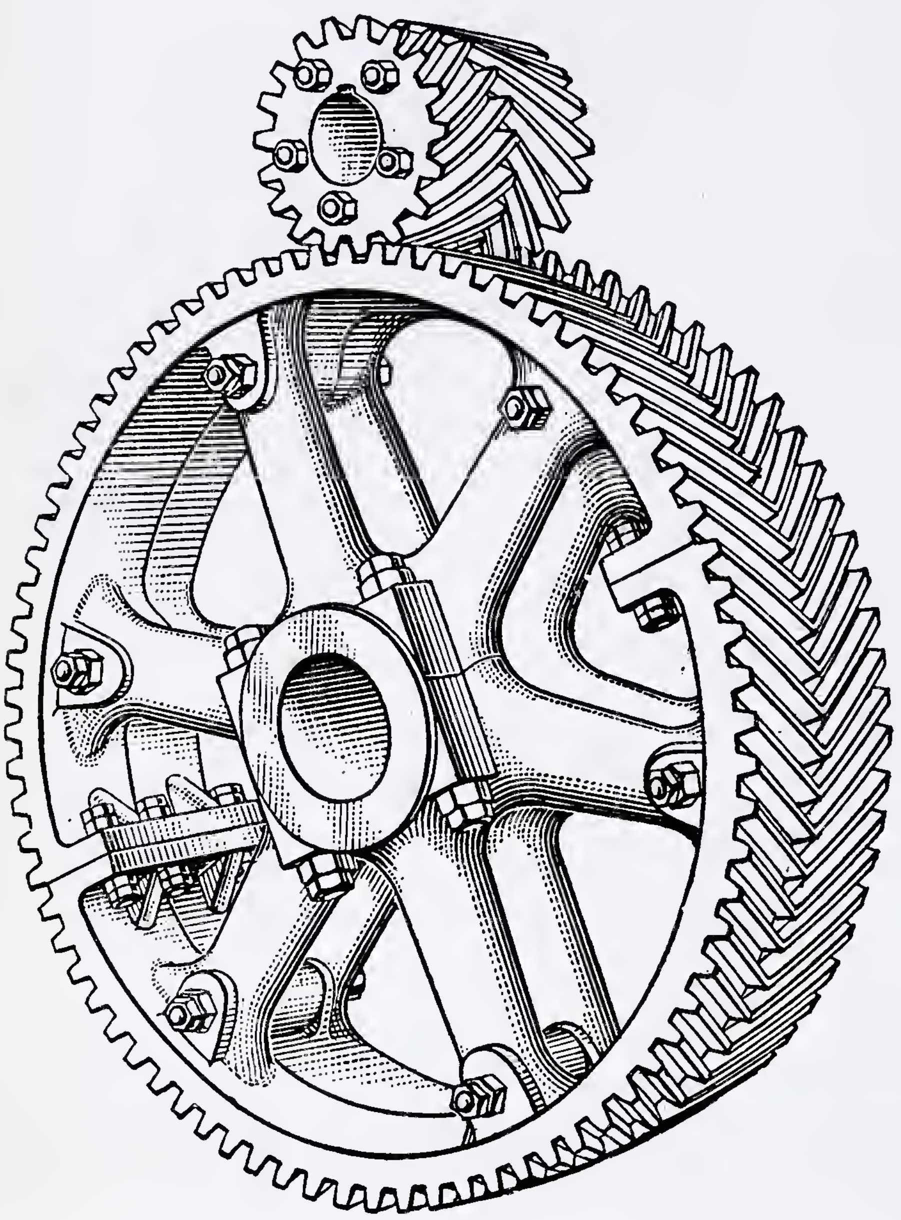


FIG. 11

16. Suction and Discharge Pipes.—The instructions given in another Section in regard to the suction pipes of reciprocating pumps, as well as the use in connection therewith when necessary of strainers, foot valves, and vacuum chambers apply also to the suction pipes of power pumps. Similarly, the discharge pipe with its accessory gate valve, check-valve, vacuum chamber, and priming pipe is the same in power pumps as in reciprocating pumps.

A power pump requires a relief valve to be attached to the discharge line near the pump, and between it and any shut-off valve. The object of the relief valve is to prevent the accumulation of pressure in the discharge pipe of such an amount as to cause damage to the pump or its attachments or the burning out of the motor. Thus, if the main discharge valve should

be closed while the pump is working, or if foreign material should lodge against the check-valve or main-valve seats, or in the elbows, and so gradually close the water passage, the pressure on the discharge side of the pump might become so great as to cause the pump to break. However, if a relief valve of ample size, that is, a valve whose opening has an area of not less than one-half that of the discharge pipe, is installed in the discharge line and this valve is set to open at a pressure but slightly above the normal working pressure of the pump, it will open automatically and reduce the pressure before the danger point is reached. A very substantial form of relief valve ordinarily made for pressures up to 50 pounds per square inch is shown in Fig. 12. This valve can be provided with a spring of such resistance that it will not open until the pressure reaches 300 pounds per square inch. A pressure gauge, such as that shown in Fig. 13, attached to the pump is of service in indicating the pressure on the plungers, and any defect in the valves, which would result in leakage and slip, may also be detected by the unusual variation in pressure. The graduations on the outer circle of the dial indicate the pressure in pounds per square inch, and those on the inner circle indicate the corre-

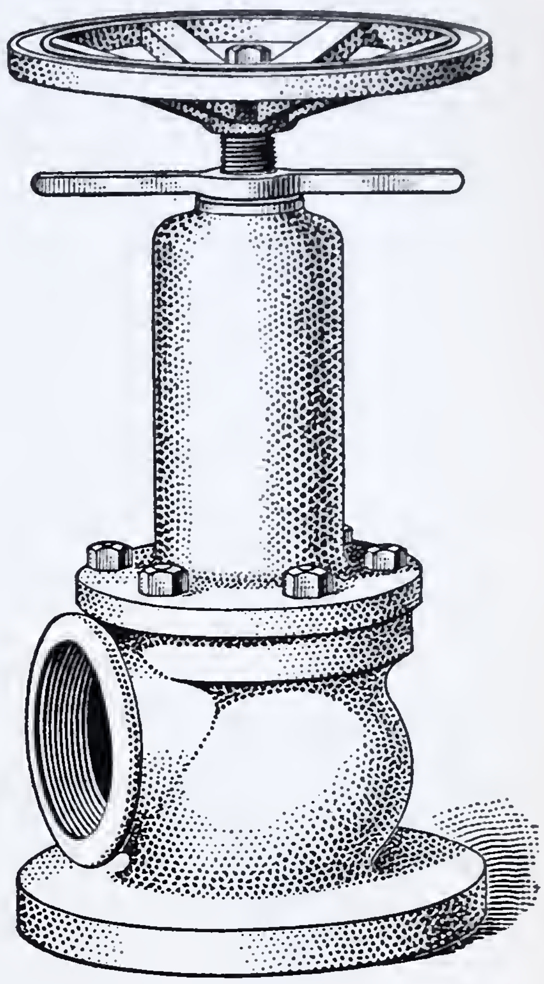


Fig. 12

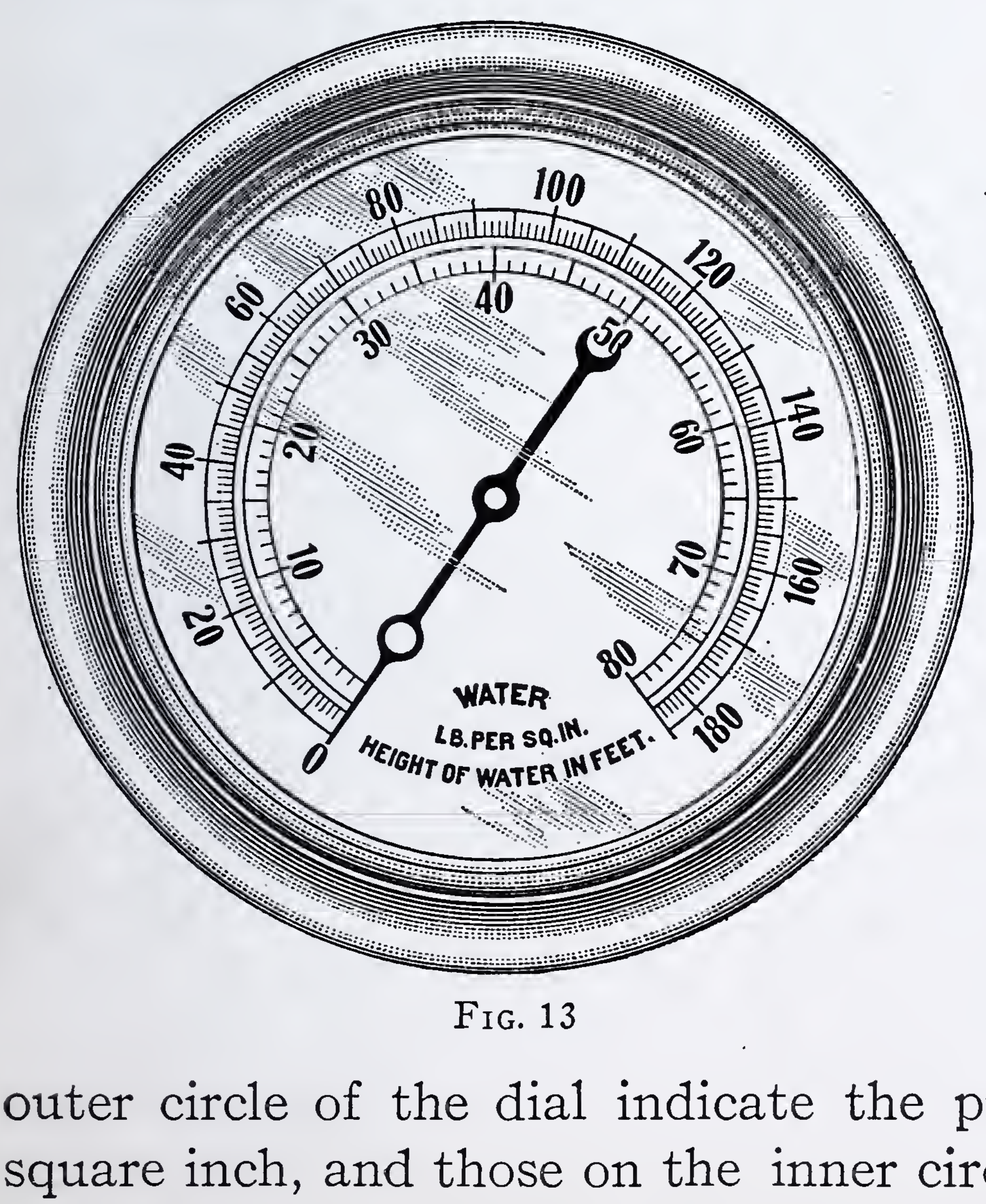


Fig. 13

sponding head of water, in feet. The use of such a gauge is often of great assistance in determining the friction head in a discharge line that is too small or of exceptional length.

17. Starting Power Pumps.—In starting a power pump, it is necessary first to see that the discharge line is open to the atmosphere and closed from the column pipe, and that the pump is free to operate in its normal cycle, so far as its moving parts are concerned. The suction pipe must be clear of obstructions and immersed to a sufficient depth in the sump to prevent air being drawn into the pipe. The drains on the suction side of the pump should be closed and those on the discharge side opened. The power should be turned on gradually and the pump run at a slow speed until water is passing through it and to the atmosphere through the valve provided for that purpose on the discharge pipe near the pump. After this circulation is thoroughly established, the next thing to do is to divert the discharge into the regular discharge pipe, close the atmospheric discharge opening, and increase the power until the pump is brought up to normal speed.

18. Power-Pump Sizes.—The sizes of power pumps are expressed by giving the diameter and stroke of the water cylinders, the number of cylinders, and by stating whether the pump is single- or double-acting. Thus, a $10'' \times 16''$ triplex double-acting power pump is a power pump with three cylinders, each 10 inches in diameter, with a 16-inch stroke, and which discharges water on both the forward and return strokes.

The piston speed of power pumps is the same as that of steam pumps under the same working conditions, and the size of the water cylinders is calculated in the same way as for steam pumps. The horsepower required for driving power pumps is also calculated in the same way as for steam pumps, but the dimensions of the driving mechanism will depend on the design of the source of power which, as previously stated, may be a steam engine, a gasoline motor, or an electric motor.

ELECTRICAL EQUIPMENT OF MINE PUMPS

19. Pump Motors.—Semienclosed motors, of the type shown in Fig. 14, are commonly used on the smaller sizes of mining pumps. These can be made completely enclosed by slight changes in the end shields *a* and *b* carrying the motor gearings, but it is, in general, not advisable to enclose completely a motor designed for continuous running, since the capacity is greatly reduced by the heat generated. The normal increase in temperature when running will protect the motor from dampness; but when idle, it would be an advantage to enclose the motor completely in a casing.

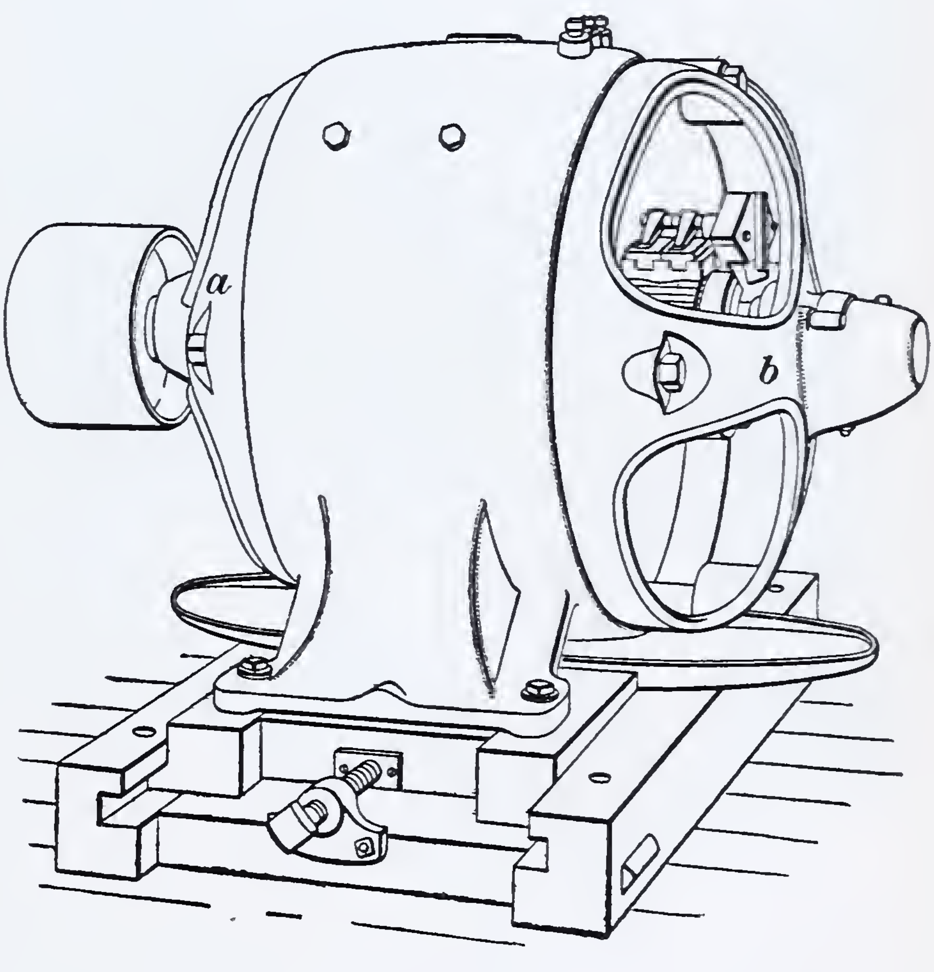


FIG. 14

Electric motors for driving power pumps should operate at as low speeds as possible, in order to reduce gear-speed. This will reduce vibration, which will be an advantage both as regards wear of gearing and bearings, and in sparkless operation of brushes and commutator. Leading electrical makers now rate two speeds for standard motors of sizes between 1 and 100 horsepower and the slow-speed type should always be chosen for geared pumps. A standard speed for 10-horsepower 230-volt motors is 650 revolutions per minute in the slow-speed type and 1,375 in the moderate-speed type. For 50-horsepower motors, of the same types, the speeds are 515 and 840 revolutions per minute. For small pumps, it is sometimes an important advantage to reduce the speed of motors below the standard. This may be done in the case of a 250-volt plant by selecting a motor of double the working voltage, and connecting the fields in multiple; the motor will

then develop one-half rated power at half speed. For example, a 5-horsepower, 500-volt, four-pole motor, operating normally at 1,000 revolutions per minute, will, with fields connected two in series on 250 volts, develop $2\frac{1}{2}$ horsepower at 500 revolutions per minute.

On many of the smaller pumps, running at the higher speeds, armature pinions are of rawhide; these are practically noiseless, and relieve the wear on the motor bearings and the driven gear considerably. They must usually be protected from rats by suitable guards, and duplicates should be kept on hand, since they are often of special dimensions.

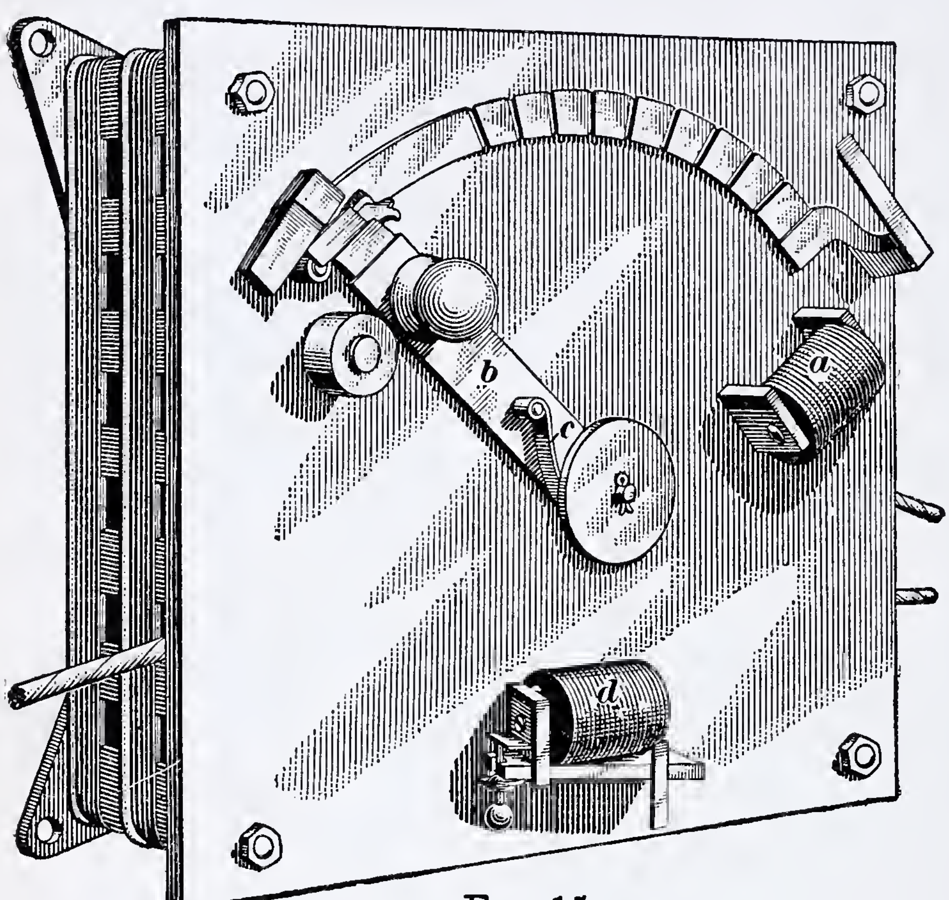


Fig. 15

20. Rheostats for Mine Pumps.—Considerable trouble is sometimes experienced with electrically driven mine pumps that are supplied from haulage or machine circuits, as it is necessary for an attendant to start the pump after each interruption from short circuit or overloads. It sometimes occurs, also, that the automatic rheostats commonly used to avoid this difficulty fail to release or they stick on some intermediate resistance point, and unless an attendant is present there is liability of damage to the rheostat or the pump motor when current is again turned into the line.

Fig. 15 shows a standard starting rheostat, with a no-voltage and an overload release. The magnet spool *a* is in series with the shunt field of the motor, and on opening the circuit this magnet releases the resistance arm *b*, which is returned to the off-position as shown, by a coiled spring *c*. The over-load-release device *d* is shown below the arm. At a previously determined overload, say 25 or 50 per cent. of the normal

load, the armature of the small magnet *d* is raised and cuts out the release-magnet spool *a* above, allowing the resistance arm *b* to swing back in the same manner as before described, cutting in resistance and finally cutting off the current.

21. An improved rheostat, which is self-operating when a current is thrown on, and which may therefore be operated from a distance, is shown in Fig. 16. A solenoid magnet *a* actuates the resistance arm *b*, and when the current is thrown on, pulls the arm around or against the resistance of a dashpot *c*, which may be adjusted to give any desired speed of starting. On interruption of the current, the arm returns to the off-position, as in the standard rheostat, and these rheostats are also made in combination with an overload circuit-breaking device. With self-oiling bearings on motor and pump, these rheostats reduce the attention required to a minimum, and one man or boy is enabled to attend to a number of pumps, or to visit them in connection with other work.

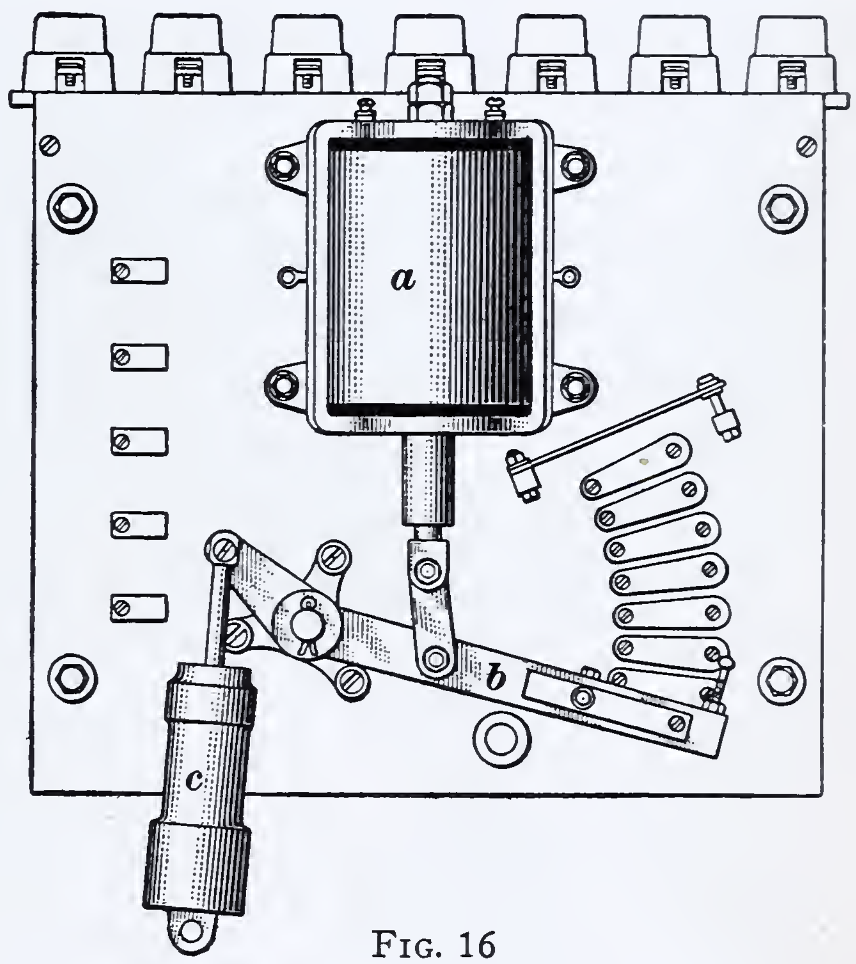


FIG. 16

22. Wherever possible, the rheostat and fuse boxes should be mounted on the iron frame of the pump or motor, insulating washers and bushings being used to avoid grounding the rheostat frame. Where they cannot be so mounted, hardwood plugs may be driven into the rib so that they project about 4 inches, and the rheostat and fuse boxes may be fastened to them. A switch should be placed to cut out the rheostat and fuses when desired; and where the current exceeds 50 amperes, some form of automatic circuit-breaker is advisable. Fuses, circuit-breaker, or overload-release rheostat should be set to operate at a small margin above full-load

current, to afford protection from any serious overload. Twenty-five per cent. overload is usually sufficient to start the pump against full head.

23. Precautions Necessary with Electrical Driven Mine Pumps.—It is the prevailing tendency, where electricity is used in mining, to locate responsibility for troubles of various kinds, often unjustly, on this, the least understood and most mysterious force employed. It is especially important, therefore, that every precaution be taken to minimize any possibility of accident from electrical causes, either by shock or by fire. While bare wires are very generally employed for feeders on the headings, it is advisable that the wires leading to the pump should be well insulated, since even a slight shock, received by the attendant working on or near the pump, may result in serious injury by his falling against it. Ordinarily, it is impossible to provide a dry wooden platform about the pump, on account of the dampness of the mine, and in some mines this is further objectionable on account of the fire risk. A small stool, with insulator pins and glasses for legs, forms a safeguard and should be used whenever it is necessary to adjust or change brushes or to work on the commutator while it is running.

Small electric pumps are often placed in mines on a wooden floor, in a frame pump house, and the attendant sometimes has a bunk or seat with straw mattress, where he reclines and smokes. Oil and waste are strewn around, the fuse boxes and rheostat are probably fastened to a board placed against inflammable material, and there is often a draft through the place. It is hardly surprising that a mine fire should originate at this point.

In comparison with a case such as the above, the following precautions are advised in placing an electric pump: If possible, the pump should be placed in a special room excavated for this purpose; but where a break-through or passage between two rooms must be used, it should be closed with brick or stone and not with wooden brattice. If located in a damp place, timber foundations may be used, but a frame structure

surrounding it, or wooden floor, is not advised. However, where the location is temporary and the expense of a brick or stone pump house is not justified, a plain wooden structure may be built, with at least 6 feet clearance from the motor end, and 2 feet from the pump proper. This may be lined, as an additional precaution, with sheet iron or tin. The use of excelsior or straw in cushions or mattress should not be permitted, and any seat or bench should be located not less than 6 feet from the motor.

Matches should not be left in or around pump houses, nor illuminating oil kept therein. Lubricating oils should be carried to the pump house in small quantities, ordinarily not more than a day's supply, and this should be kept in closed cans. Cotton waste when saturated with oil is liable to take fire spontaneously and should be kept in tight cans until taken out of the mine, and should not be thrown about on the floor.

The pump house should be well lighted by incandescent lamps. Where lamps are used in series, at least two circuits should be run, since one burn-out will extinguish an entire series.

The transmission wire for the pump should be carefully placed on glass or porcelain insulators, and extra care should be taken that it does not rest on the pump frame or mine timbers. The line should be taken as directly as possible to the switch and fuses, so that the latter may be protected against a series-ground or a short circuit beyond.

It is a wise precaution to tap the discharge pipe of the pump with 1-inch connections and keep in the pump house a length of hose with a nozzle, for use in case of fire. A precaution of this kind may save several thousand dollars.

CENTRIFUGAL PUMPS

TYPES AND APPLICATION

24. Development of the Centrifugal Pump.—The handling of water by machinery operating on the principle of centrifugal force was first accomplished in 1703, when Papin, a French engineer, embodied the essential features of the present-day centrifugal pump in a successful machine. From 1846 to 1851 considerable development in pumps of this style was made, but successful operation could not be secured at heads exceeding 20 to 30 feet. Since that time, steady advancement in scientific knowledge and improvement in foundry and shop practice have developed this type of pump to such a degree that it can handle large quantities of water not only at low heads but at heads up to 2,000 feet, or about 850 pounds pressure per square inch, and this performance is accomplished at an efficiency equal to, or under certain conditions, superior to, that obtained by any other method of pumping water.

The earliest centrifugal pumps were failures because of low efficiency; but with the modern general tendency toward high rotative speeds in almost all prime movers, the power consumption of recent centrifugal pumps of small capacities compares favorably with that of the best geared power pumps, and for the larger capacities they have a decided advantage over reciprocating pumps, particularly where the working head does not exceed 150 or 200 feet.

25. Classes of Centrifugal Pumps.—There are two general classes of centrifugal pumps, which are known respectively as *volute pumps* and *turbine pumps*. A simplified cross-section of a volute pump is shown in Fig. 17 (*a*). The pump, as shown, consists essentially of two parts, which are the *impeller* *a* fastened to the shaft *b*, and the casing *c* having a spiral form

either conforming to or approximating the curve known as the *volute*, whence the pump derives its name. The cross-section of the passage into which the water is discharged by the impeller, while always circular, gradually increases in area as the discharge opening of the pump is approached. The pump casing may have a form different from that of a volute; it may even be circular, but experience has shown that with a casing differing much from the volute form the pump will be very inefficient. The impeller is rotated from any convenient source of power by belt, chain, or rope drive, or is direct-connected to an engine or electric motor; it has a number or blades, or vanes, curving backwardly in relation to its rotation, and is also called by various other names, such as *rotor*, *runner disk*, *runner*, *flyer*, or *fan*. The water inlet is at the center of the impeller, and water entering the revolving

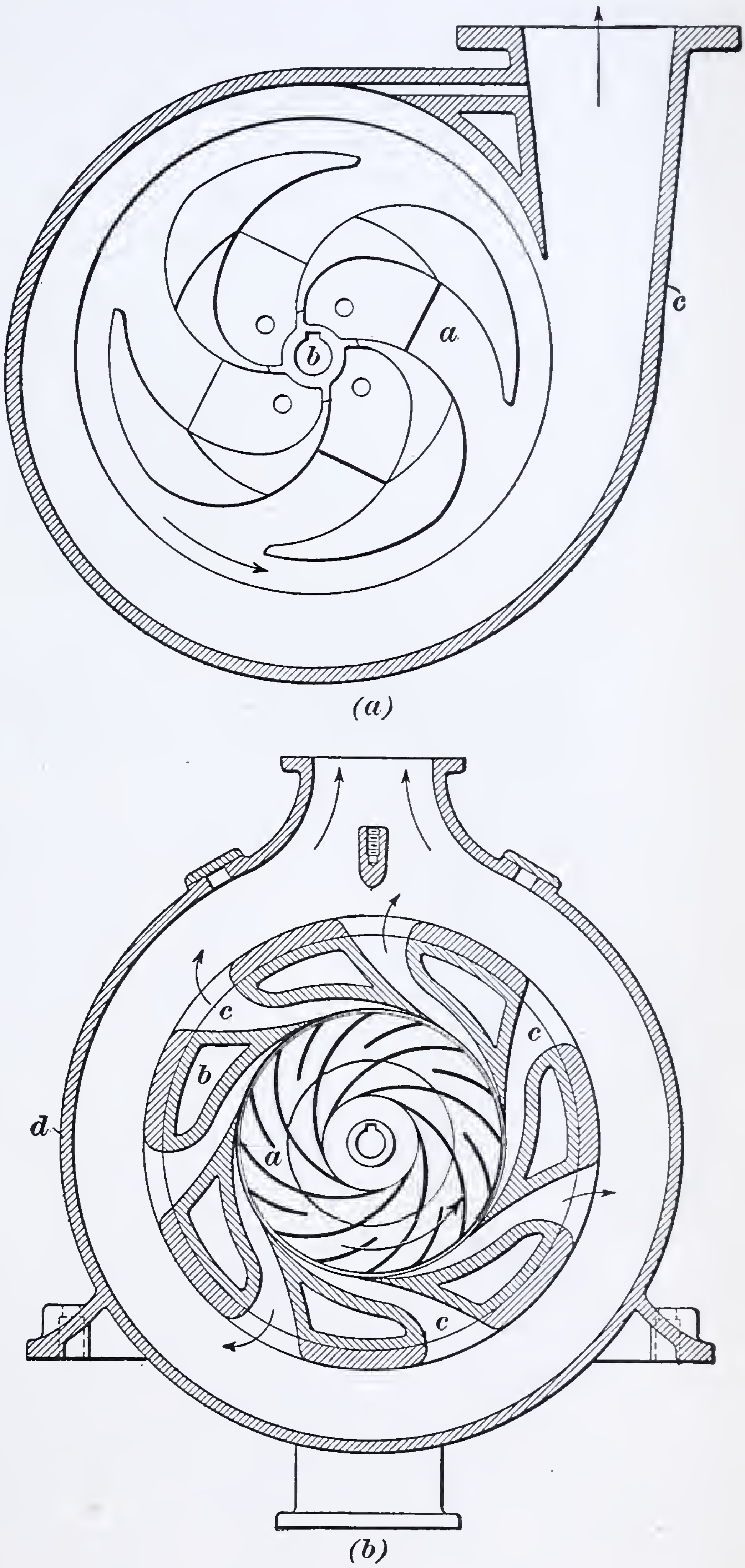


Fig. 17

impeller is carried around with it and at the same time travels outwards toward the periphery of the impeller, whence it is discharged at a high velocity into the volute of the casing and passes onward to the discharge outlet. The velocity of the water leaving the impeller is gradually reduced as the area of the volute part of the casing increases, and the kinetic energy of the moving body of water is thus transformed into pressure.

The turbine pump, shown in Fig. 17 (*b*), consists essentially of three parts, which are the impeller *a*, a diffusion ring that carries the diffusing vanes *b* which form a series of passages *c* increasing in area outwards from the edge of the impeller, and the casing *d* which may be circular, as shown, or may be composed of two volutes. The purpose of these vanes is the same as that of the volute, namely, to reduce the velocity and increase the pressure of the water leaving the impeller.

26. Centrifugal pumps of both the turbine and the volute types are classified as *single-suction* and *double-suction pumps*. In a single-suction pump the water enters at one side of the impeller, and consequently exerts a side thrust on the impeller, which must be resisted by suitable means; in a double-suction pump the water enters both sides of the impeller, and consequently there is no side thrust. For pumping against fairly low pressures centrifugal pumps are usually made *single-stage*, which means that the pump has only one impeller; for pumping against high pressures, centrifugal pumps are made *multistage*, which means that the pump has a number of impellers which operate in succession on the liquid pumped. In a multistage centrifugal pump the discharge of the first impeller passes under pressure to the inlet around the center of the second impeller and leaves its periphery at an increased pressure, passing thence to the center of the third impeller, and so on. The number of stages used depends on the final pressure to be pumped against; sometimes ten impellers are employed. Single-stage and multistage centrifugal pumps are built in both the single-suction and double-suction types.

27. Single-Stage Volute Pump.—The simplest form of centrifugal pump is the single-stage volute pump with single

suction; such a pump is shown in longitudinal section in Fig. 18. The shaft *a* is rotated by a belt that drives the pulley *b* and carries at one end the impeller *c* which is a circular disk having curved vanes or blades *d* radiating outwards from the center. These blades are of the form shown in Fig. 17 (*a*). The impeller is keyed to the shaft *a*, Fig. 18, and rotates inside the closed casing *e*, which, at its outer edge, is formed into an opening *f*, of circular cross-section and volute curvature, that leads into the discharge pipe *g*. The suction pipe is connected to the flange *h* and the water enters the pump casing at *i*.

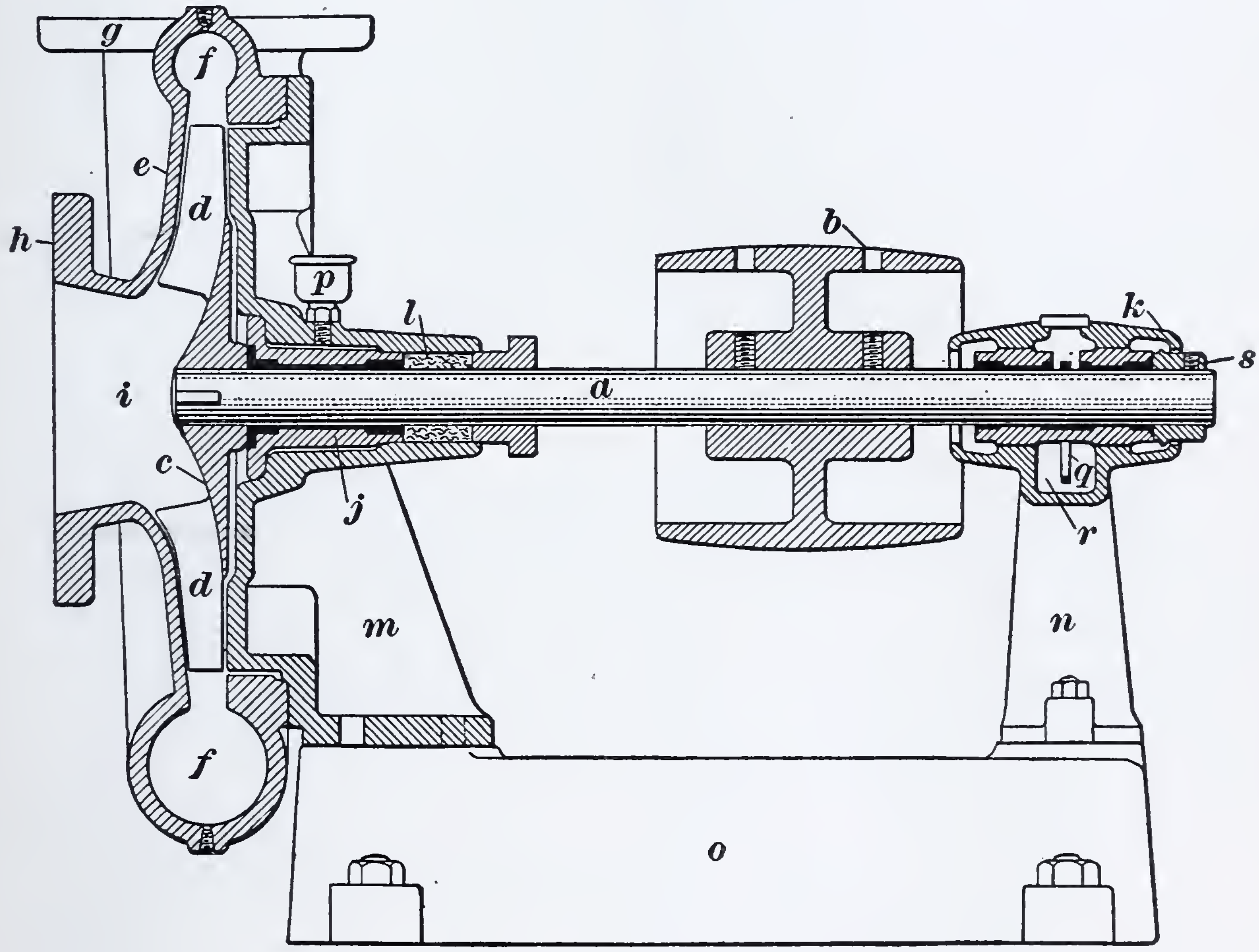


FIG. 18

Bearings *j* and *k* support the shaft *a*, and a stuffingbox *l* prevents the leakage of air into the pump chamber along the shaft. The pedestals *m* and *n* that contain the bearings are bolted to the single base plate *o*. The bearing *j* is lubricated by the oil cup *p*, while the bearing *k* is lubricated by a ring *q* that turns on the shaft *a* and carries oil up on top of the shaft from a well *r* in the pedestal.

As already mentioned, in single-suction pumps there is an end thrust on the shaft, because the pressure on the suction side of the impeller is less than that of the atmosphere outside.

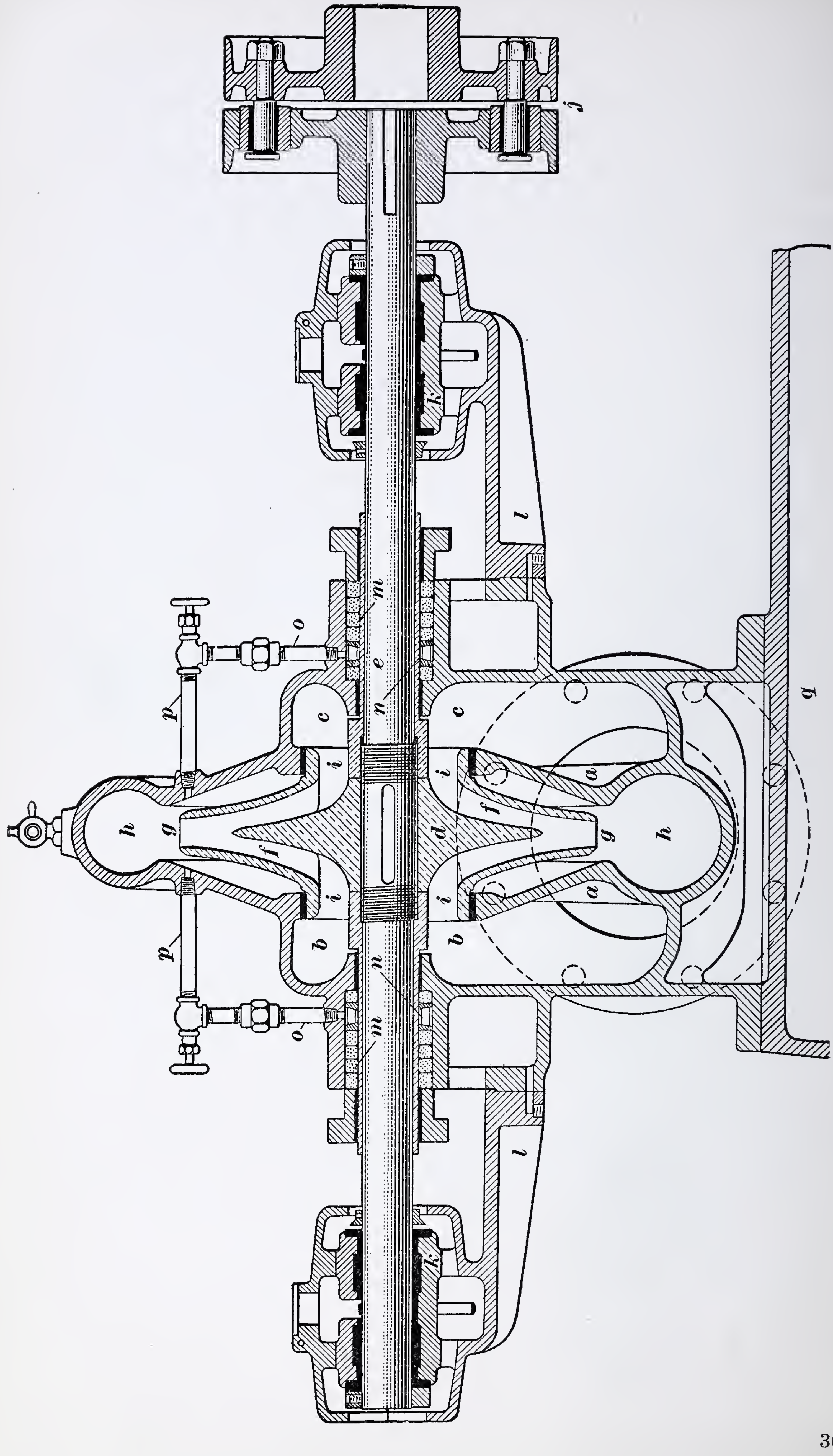


FIG. 19

To prevent endwise movement of the shaft, a thrust collar *s* is fastened to it, this collar bearing against the end of the bearing *k*.

Before the pump will operate, it must be primed; that is, the suction pipe, the chamber *i* and the passage *f* must be filled with water. When the shaft *a* is rotated, the centrifugal force set up throws the water outwards toward the tips of the vanes of the impeller and into the passage *f*. The movement of the water away from the center causes a vacuum in the chamber at *i*, and water flows up through the suction pipe into the pump to fill this vacuum. Thus a continuous flow of water from the suction pipe through the impeller into the passage *f* and out of the discharge *g* is produced.

28. Double-Suction Volute Pump.—Another form of single-stage volute pump is shown in section in Fig. 19. It differs from the pump described in Art. **27** in that it is of the double-suction type in which water flows into the impeller from both sides. The suction pipe *a* opens into the passages *b* and *c* on opposite sides of the impeller *d*. When the shaft *e* is rotated, the water in the impeller passages *f* is thrown out at the tips *g* into the passage *h*. At the same time, water from the suction chambers *b* and *c* flows into the impeller through the annular openings *i*. As previously explained, end thrust on the shaft is avoided by having the same pressure on both sides of the impeller, by reason of the double suction.

This pump is intended to be driven directly by a motor connected to the shaft *e* by the coupling *j*. The bearings *k* are both external, and are carried by brackets *l* bolted to the main frame of the pump. As there are two suction chambers, a stuffingbox *m* must be provided at each side of the casing to prevent the inward leakage of air. In each stuffingbox is a hollow metal ring *n*, known as a water-seal cage, the interior of which communicates with the discharge side of the pump by way of the pipes *o* and *p*. Water is thus allowed to enter the water-seal cage and surround the shaft at that point, thereby preventing air from entering the pump by leakage along the shaft. This arrangement is called a water seal, and is

discussed further on. The pump is supported by a rigid base *q* that also forms a support for the motor by which the pump is driven. Pumps of this type are suitable for heads of 150 to 175 feet and they are readily designed for capacities exceeding 5,000 gallons per minute.

29. A perspective view of a double-suction single-stage volute pump with the top half of the casing removed is shown in Fig. 20. This pump differs only in minor details from that shown in Fig. 19, and is illustrated here to bring out more

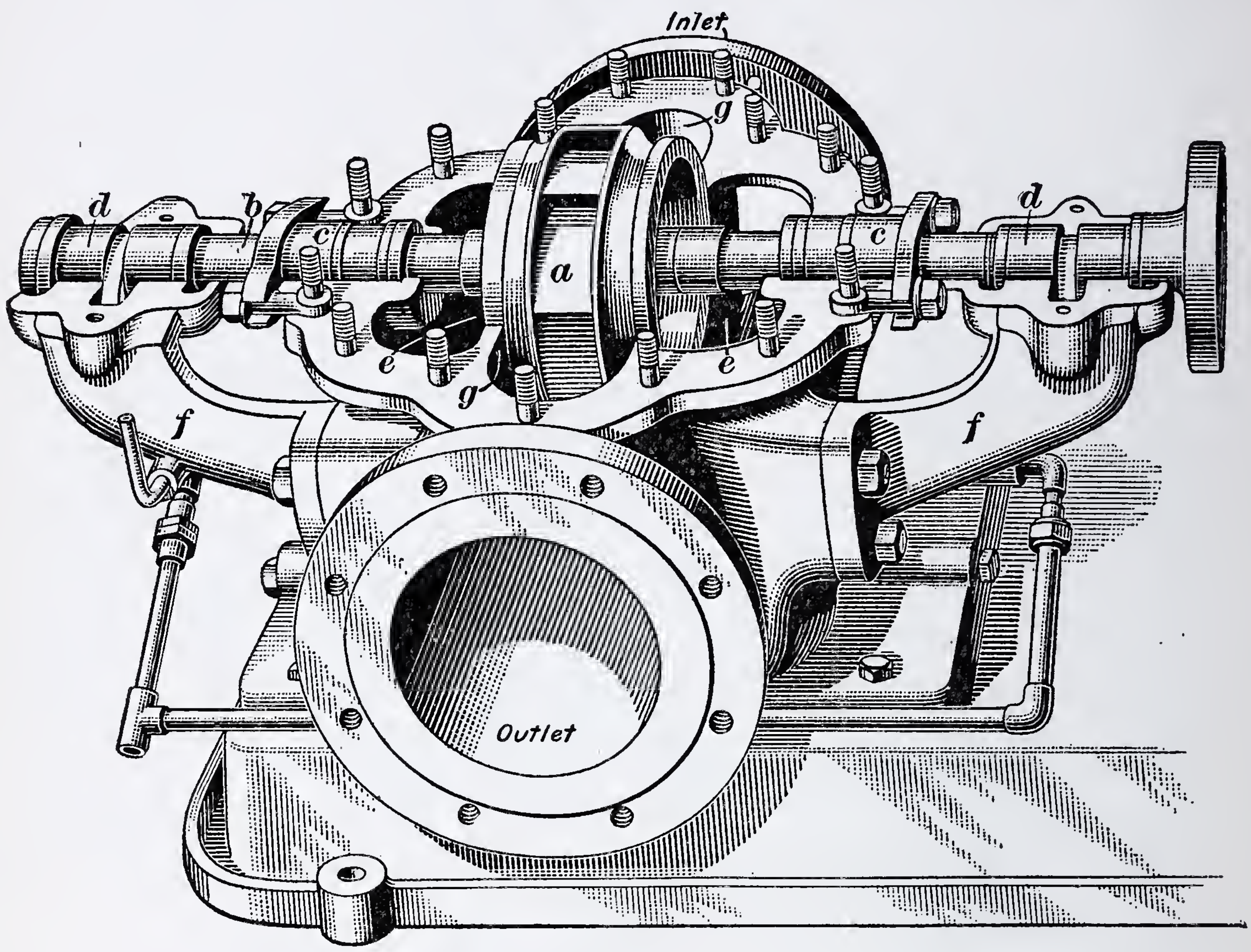


Fig. 20

clearly, than is possible in a sectional view, the actual appearance of the inside of a centrifugal pump. The impeller *a* is a bronze casting fastened to the steel shaft *b* which passes through the two stuffingboxes *c* and is supported by the two ring-oiled bearings *d*. These bearings are made independent of the stuffingboxes *c*, to prevent the impeller *a* and the shaft *b* from getting out of alinement through wear. and also to prevent the lubricant from entering the pump chamber *e*. Within the latter the shaft *b* is surrounded by renewable bronze sleeves. These sleeves are screwed into the impeller, so as to prevent the liquid that is being pumped from coming in con-

tact with the shaft. To prevent entrance of air around the shaft and leakage of a cold liquid, cupped-leather packing rings, which are set out or held tightly to the shaft by the pressure of the liquid, are placed in the stuffingboxes. When pumping a hot liquid, however, graphited flax packing is substituted for the leather packing rings.

The impeller takes in water from the suction, or inlet, chambers *e* at the center, on both sides of its hub, thus preventing end thrust; the water enters these two chambers from the single water inlet. The water is discharged through the passages between the curved vanes, or blades, to the periphery and into the outlet, or discharge, chamber *g* terminating in the delivery pipe.

The brackets *f* that carry the bearings *d* form oil reservoirs at their outer ends, which reservoirs contain oil into which dip oil rings that by their rotation with the shaft *b* carry oil up to the bearings. The inner ends of the brackets *f* form drip boxes that catch any leakage of water from the stuffingboxes; drain pipes convey any drip water to some convenient source of disposal.

Leakage from the discharge side of this centrifugal pump to the inlet side, past the sides of the impeller, is prevented by grooved packing rings, one pair of rings being carried by the impeller *a*, while a pair of stationary rings are carried by the casing. The circular grooves of the rings interlock and are a fairly close fit; the grooves form a very tortuous path for water to leak through, and owing to this tortuous path the packing rings are said to constitute a *labyrinth packing*. This form of packing is described more fully in another article.

30. Multistage Centrifugal Pumps.—When the head to be pumped against exceeds 150 feet, the multistage pump may be employed. The principle on which it operates is the same as that of the pumps already described, but, as noted in Art. **26**, it consists of two or more single-stage pumps arranged in series; that is, the water discharged from the first pump of the series enters the suction of the second pump, and so on. Such an arrangement may be made to include any desired number of

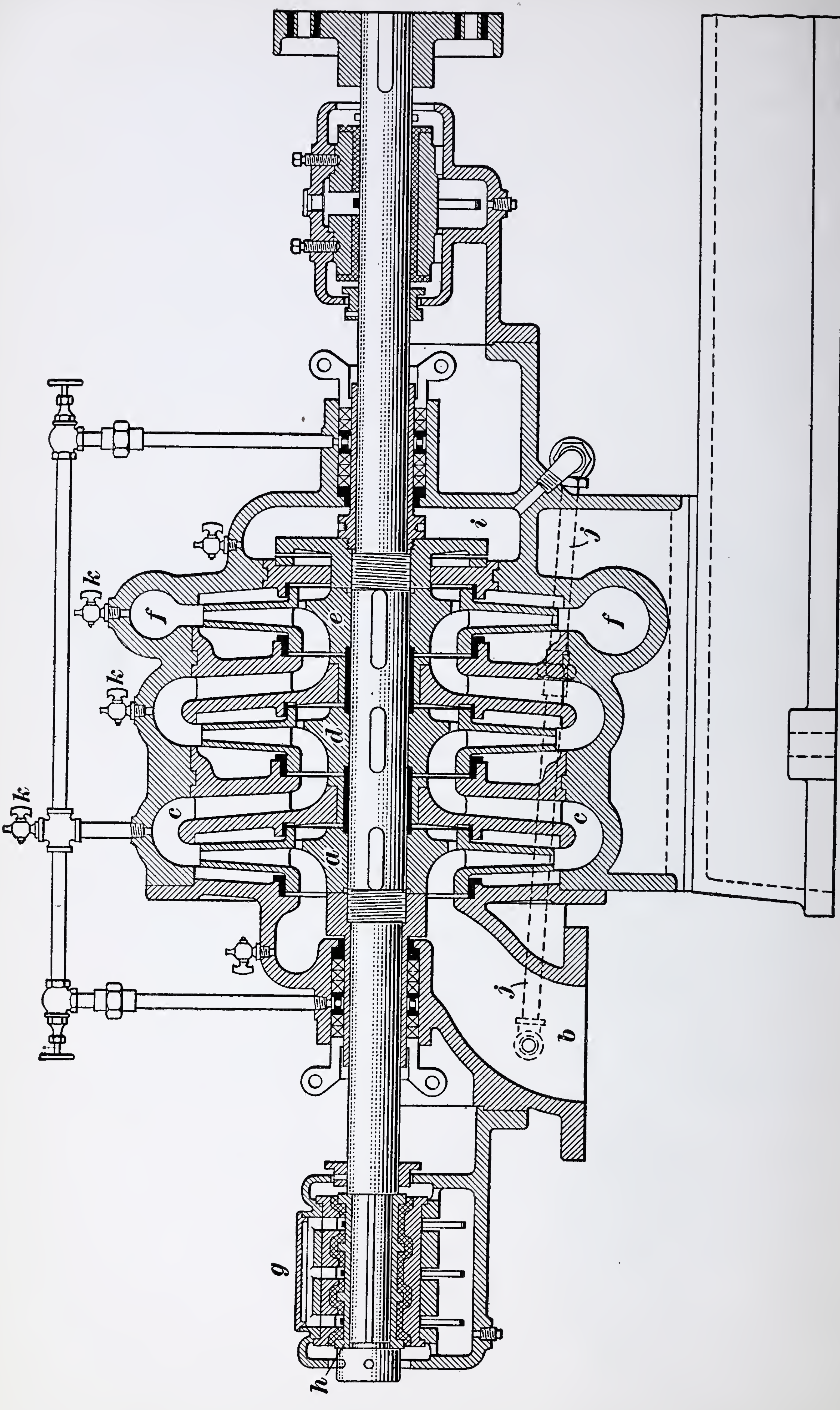


Fig. 21

pumps, or stages; but about five stages is the usual limit in practice, although more may be used.

A three-stage pump is shown in section in Fig. 21. Water enters the first pump *a* through the suction pipe *b* and is discharged into the passages *c*, by which it is led to the intake side of the second pump *d*. The discharge from this pump, in a similar manner, is led to the third pump *e*, from which it is discharged into the passage *f*, which communicates with the discharge pipe of the pump.

As the intake of each stage of the pump is at the left of the impeller of that stage, there is an unbalanced pressure that tends to move the pump shaft endwise. To prevent such movement, a thrust bearing is provided at *g*, consisting of a steel sleeve *h* fixed to the pump shaft and carrying four collars that engage with grooves in the bearing shell. At the discharge end of the pump a chamber *i* is formed in the casing, and this chamber is connected to the suction side by the pipe *j*. Any water that leaks past the final stage of the pump collects in the chamber *i* and is drawn back into the suction through the pipe *j*. Cocks are provided at *k* to enable air to be drawn out of the pump when it is being primed preparatory to starting.

31. When two or more centrifugal pumps are arranged in series, so as to form a multistage pump, the head against which the combination will work is equal to the sum of the heads produced by the individual stages of the pump; in other words, if three pumps, each capable of working against a head of 60 feet, are combined into a three-stage unit, the final discharge from the last stage will have sufficient pressure to overcome a head of 180 feet. In the multistage pump, the several impellers are keyed to one shaft and rotate at the same speed. Usually, the pump casing is in two parts, bolted together in a horizontal plane through the axis of the shaft. This construction, which is illustrated in Fig. 20, permits the upper half of the casing to be removed readily, so that the interior may be inspected or repairs made. If the pump is of large capacity and must work against a high head, it is advisable to use a flexible coupling between the pump and the motor.

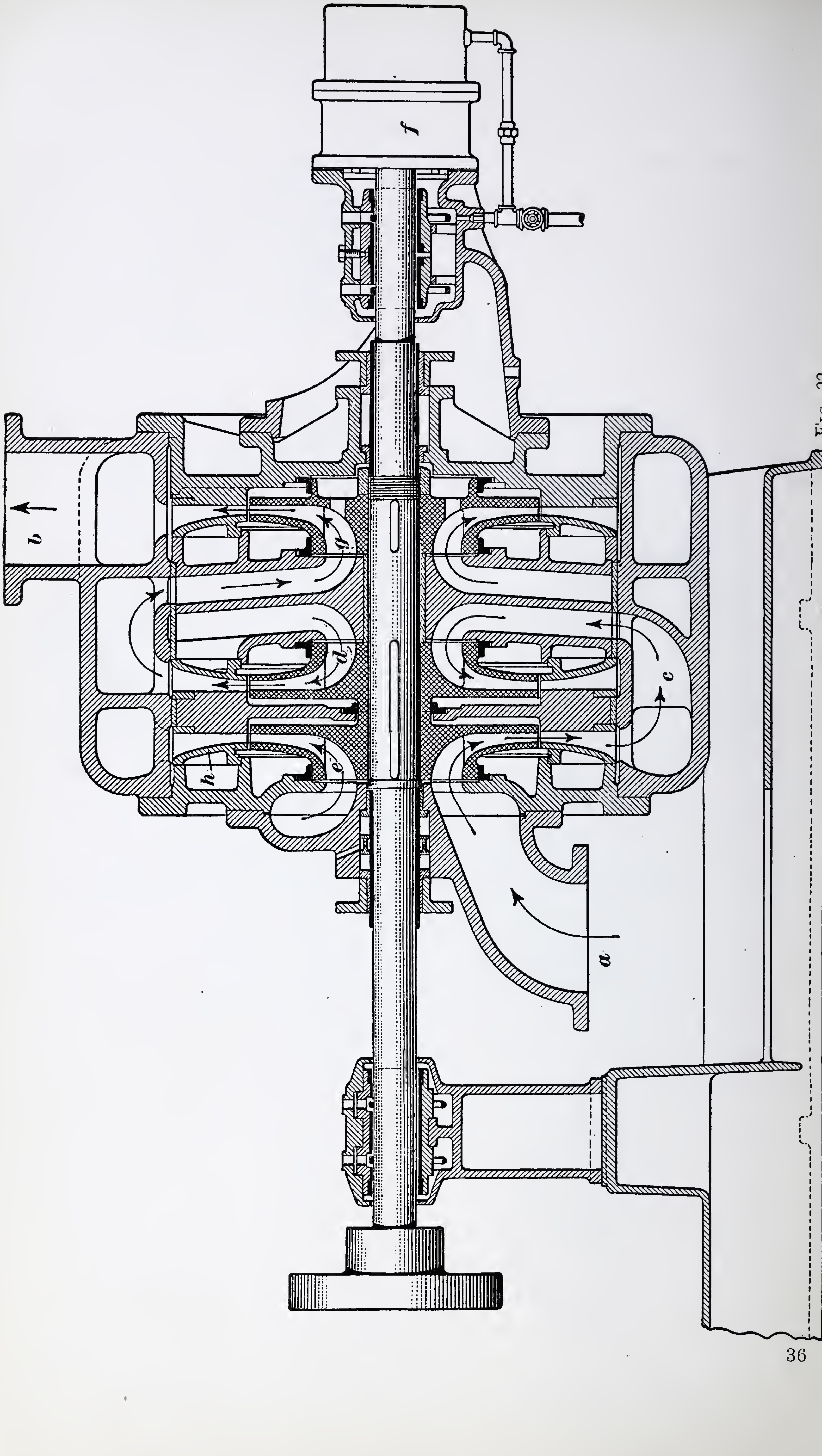


Fig. 22

32. Three-Stage Balanced Pump.—A three-stage balanced pump of the turbine type is shown in Fig. 22. The suction pipe is connected at *a* and the water in passing through the successive stages follows the general direction of the arrows, being discharged at *b*. It will be seen that the transfer passage *c* is so arranged that the water enters the second impeller at *d* on the side opposite that on which it enters the first impeller *a*. As a result, the end thrust produced on these impellers is balanced, and the bearing provided at *f* needs to take care of the thrust of only the one unbalanced stage *g*. At *h* is a

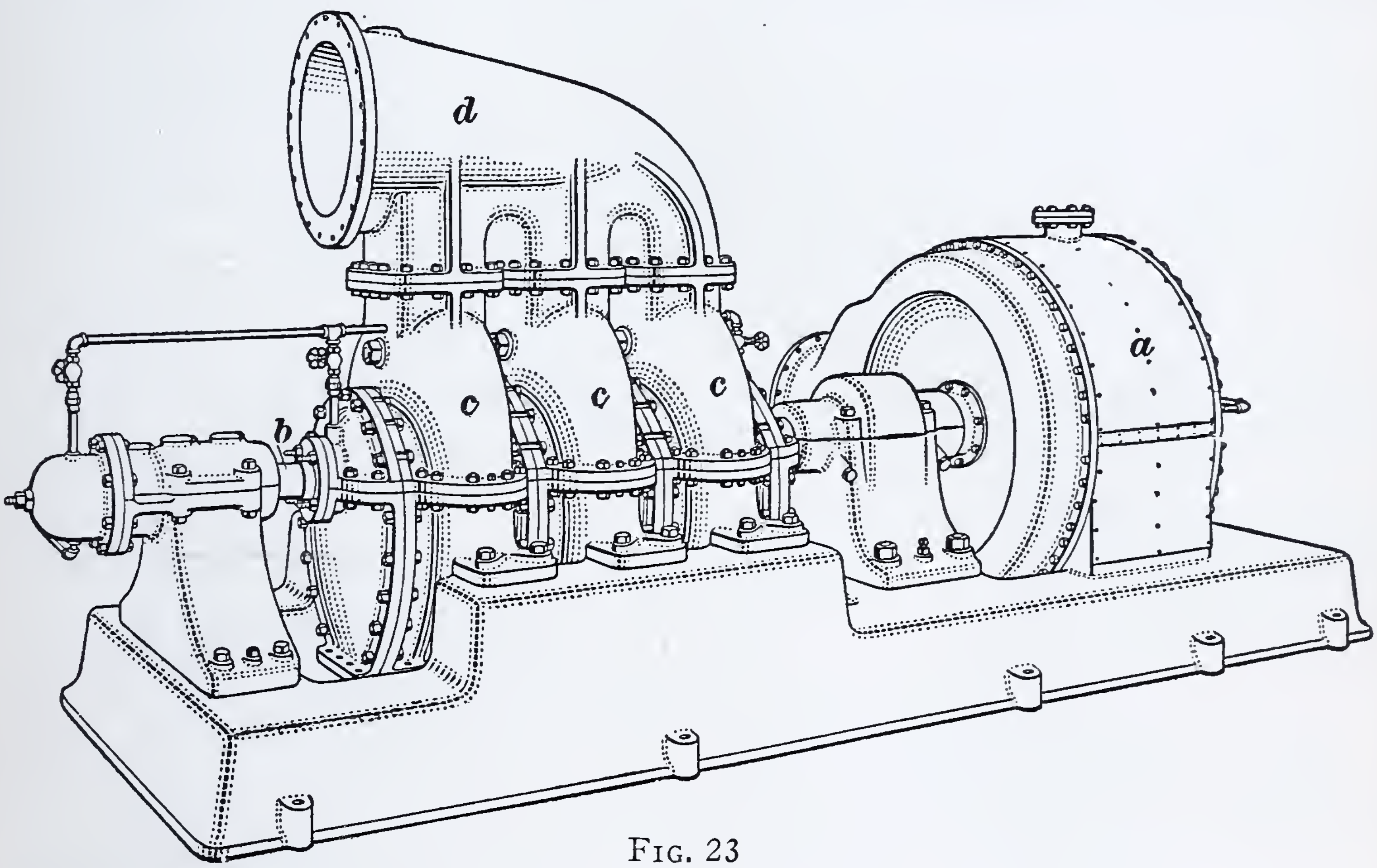


Fig. 23

diffusion ring, the use of which instead of a volute casing to convert the kinetic energy of the water into pressure, was explained in Art. **25.**

33. Pumps in Parallel.—When a very large quantity of water is to be pumped to a moderate height by a centrifugal pump, this may be done by a single very large pump driven by a steam engine or other slow-speed prime mover, or it may be done by using a number of small pumps all connected to the same suction pipe and all discharging into the same discharge pipe; pumps thus connected are said to be *in parallel.* When small pumps are chosen, they are usually driven by a steam turbine or directly by an electric motor, which two types of prime movers are essentially high-speed machines.

Sometimes several separate pumps in parallel are arranged to be driven simultaneously by the same source of power; in other cases several impellers are combined into a single casing, all impellers being in parallel. Pumps constructed in the manner last named are called *multi-impeller centrifugal pumps*, and are usually of the single-stage type. The chief advantage of using small pumps in parallel, rather than a single large pump and slow-speed source of power, is the low cost of the installation.

A multi-impeller single-stage double-suction turbine pump is shown in Fig. 23. The steam turbine *a* is direct-connected to the impeller shaft *b*; the three water outlets of the pump casing are connected to the discharge manifold *d*. There are three water inlets, one for each impeller, which are connected to a single inlet manifold not shown in the figure.

In the series arrangement of pumps the volume is fixed while the head varies according to the number of pumps, whereas in the parallel arrangement the head is fixed and the volume varies with the number of pumps.

DETAILS OF CONSTRUCTION

34. Balancing Elements.—The thrust or tendency of the shaft to move endwise in a centrifugal pump is caused by the unbalanced pressure created in various ways on the impeller. This unbalanced pressure is in part due to the vacuum existing on the inlet side of the impeller, and may be increased by the high pressure on the discharge side, or back wall, of the impeller. Simple thrust bearings, such as those mentioned in preceding articles, are used to resist this thrust, and special devices are employed to counterbalance it. One of the special devices, known as a hydraulic balancing element, is shown in cross-section in Fig. 24. To the pump shaft *a* is fixed a disk *b* having at its outer edge a shoulder *c* that matches with a similar shoulder *d* on the stationary piece *e*. An annular chamber *f* is thus formed between the two parts, and this chamber communicates by a passage not shown, with the discharge of the pump. As a result, water at the discharge

pressure enters the chamber *f* and tends to force the disk *b* and the shaft *a* towards the right. The partial vacuum formed on the left of the impeller tends to cause movement of the impeller and the shaft toward the left. Thus, these two opposing forces are balanced, and no movement in either direction results. Water escapes from the chamber *f* between the faces of the shoulders *c* and *d* and collects in the space *g*, from which it is drawn into the suction line of the pump through a connecting pipe that is not shown.

A mechanical thrust bearing is better suited for use where the water is gritty or acidulous because the hydraulic balance is a very sensitively adjusted contrivance, and the rapid wear resulting from the action of grit or acid on the finely finished surfaces interferes with its proper functioning. There are numerous mechanical thrust bearings, such as the marine type, shown in Fig. 21, which consists of a number of collars fixed to the end of the shaft, the collars working in babbitted bearings lubricated by oil rings. The Kingsbury thrust bearing consists of two hardened steel collars attached to the shaft and revolving with it, which transmit the thrust to two stationary bronze shoes. The collars are made so that when the pump is in operation they lift slightly at the end opposing the direction of rotation and admit a wedge-shaped film of oil between the surfaces under pressure. The bearing is lubricated by oil fed by a small pipe into the center of the shaft at the thrust bearing, whence it is distributed by centrifugal force to all points needing oiling. The end of the bearing is water jacketed to cool the oil which becomes heated as the bearings revolve.

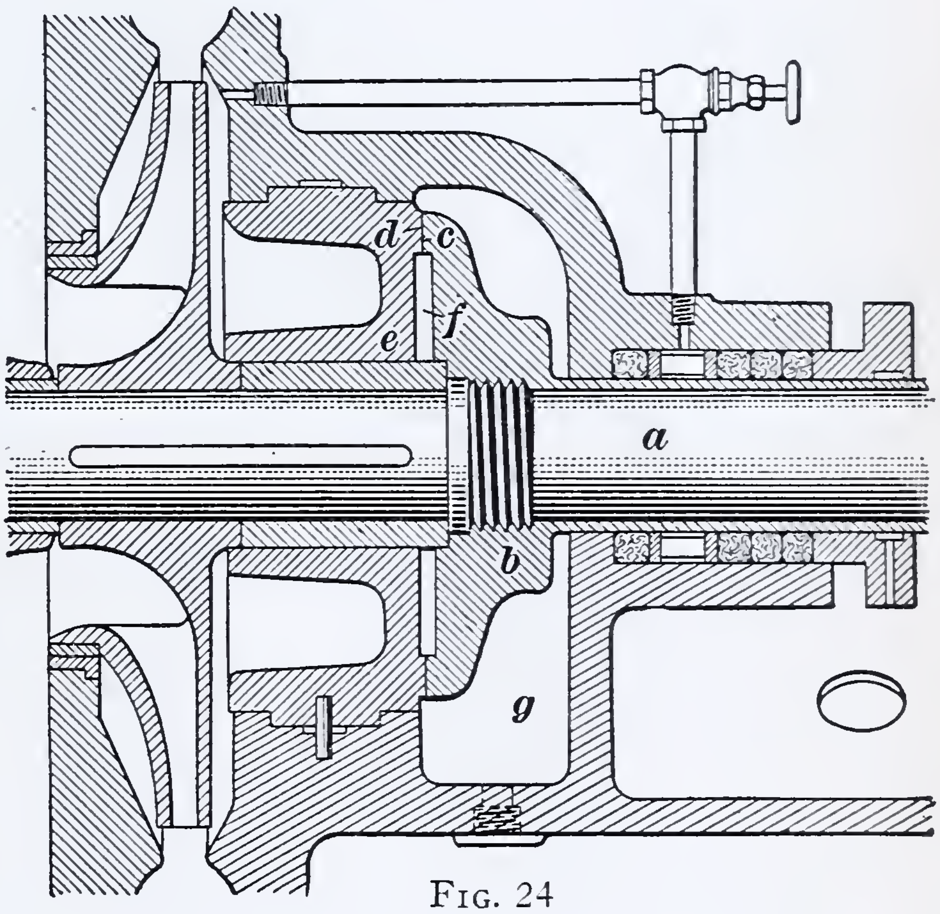


Fig. 24

35. Water Joints.—Another factor in preserving the efficiency of a centrifugal pump is the prevention of leakage between the impeller hub and the pump casing. The proper type of joint is determined by the character of the liquid handled. Where highly viscous liquids are pumped, an ordinary joint consisting of a plain ring fitted into the case, and surrounding the hub of the impeller, is allowable, the clearance space between the two acting as an effective water seal. But where thin liquids, such as water, are to be handled, more elaborate provision must be made to maintain a proper degree of tightness at the joints.

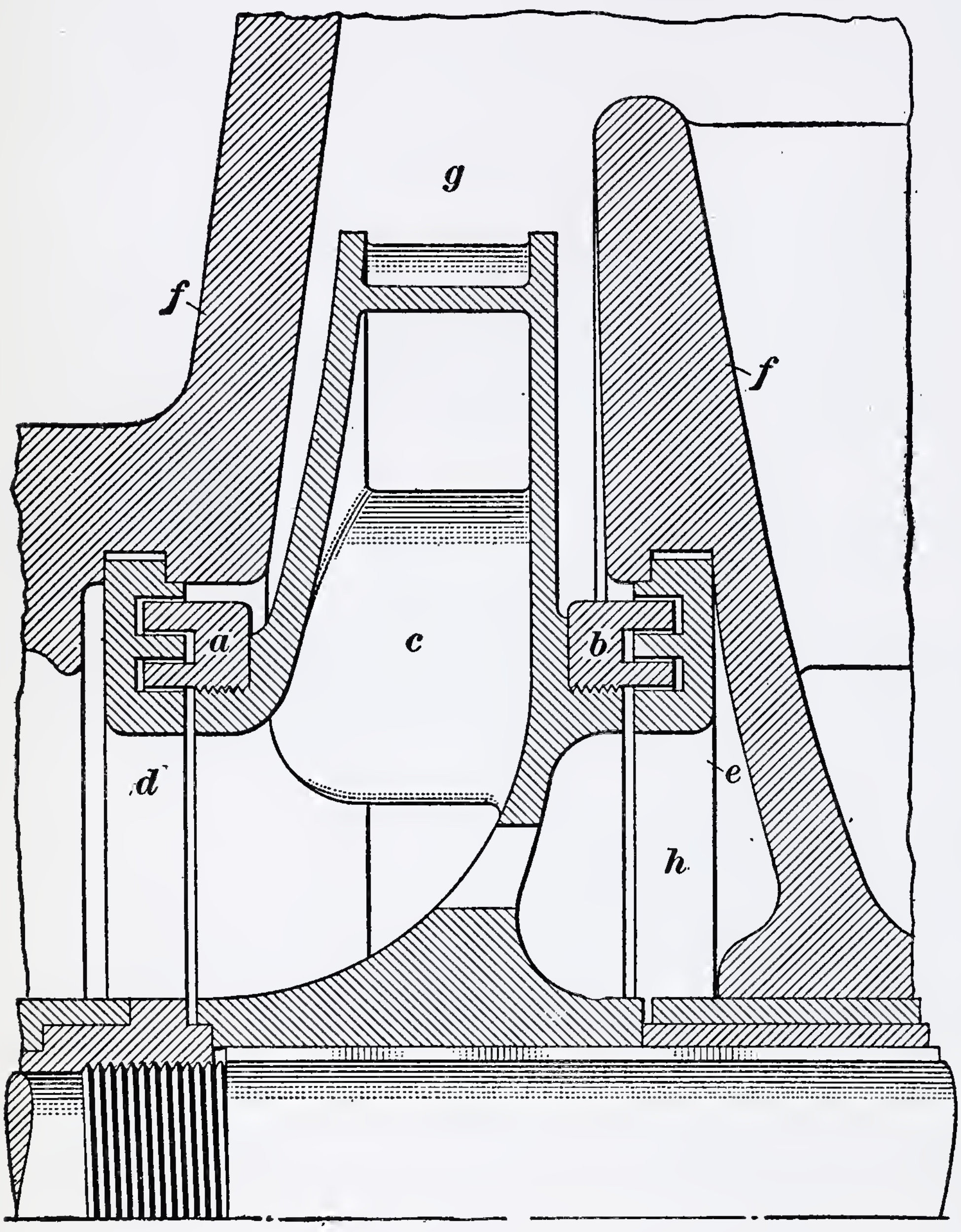


Fig. 25

The device used to prevent leakage between the rotating and the stationary parts in a high-pressure multistage pump is called a clearance ring, or labyrinth ring, as stated in Art. **29.** The use of this ring is clearly shown in the section view in Fig. 25. Two labyrinth rings *a* and *b* are firmly fastened to the impeller *c*, on opposite sides, and rotate with the impeller. The tongues in these rings fit into grooves in the stationary rings *d* and *e*, which are held securely in the casing *f*. The rotating rings do not touch the stationary rings, but fit close to them, so that any leakage that occurs between the high-pressure chamber *g* and the low-pressure chamber *h* must follow a winding path between the labyrinth rings; and when the spaces between the rings are filled with water, leakage is slow and difficult. This joint is one of the most difficult to

make and keep tight, and leakage through it cannot ordinarily be detected except when it becomes so great as noticeably to reduce the capacity and efficiency of the pump. The necessarily small clearance between the rings explains why it is imperative to have a centrifugal pump filled with water before it is started; for failure to do this is apt to result in injury through contact of the stationary and rotating rings. The thrust bearing must be kept in perfect adjustment, for a slight endwise movement of the shaft will cause damage to the rings and loss of pump capacity and efficiency.

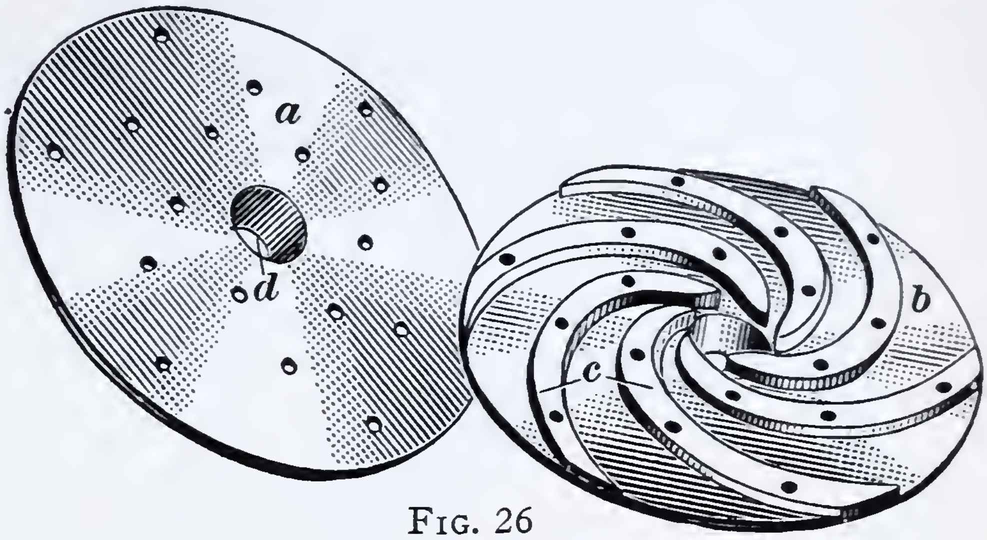
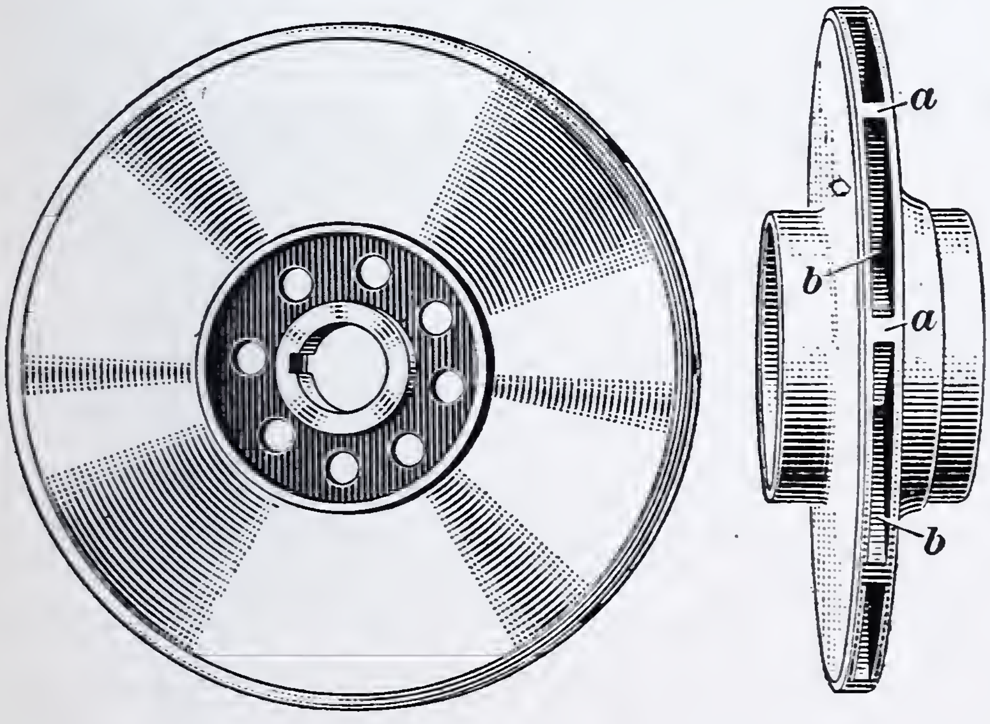


Fig. 26

36. Impellers.—The impeller is the most important element in the centrifugal pump; for on its design, construction, and finish, the efficient working of the pump depends. One form of impeller of small size is shown in Fig. 26. It consists of two parts *a* and *b*, made of brass, the part *b* being cast with curved vanes *c*. These vanes are carefully machined and smoothed, and then the two parts of the impeller are riveted together. The entire piece is finally mounted on a lathe, turned, trued, balanced, and finished all over, so that the greatest possible efficiency may be obtained. This form of impeller is of the single-suction type, because water enters from one side only, through the opening *d* in the disk *a*.



(a) (b)

Fig. 27

A one-piece impeller is shown in Fig. 27, (*a*) being a side view and (*b*) an edgewise view. It consists of two disks separated by vanes; the ends of which can be seen at *a*. These vanes form the curved passages *b* between the disks through which the water flows under the action of centrifugal force. Suction takes place on both sides, through the openings in the hubs. This type of impeller is commonly made of cast bronze. All easily accessible surfaces are machined, and those that are more difficult to reach are carefully finished by hand.

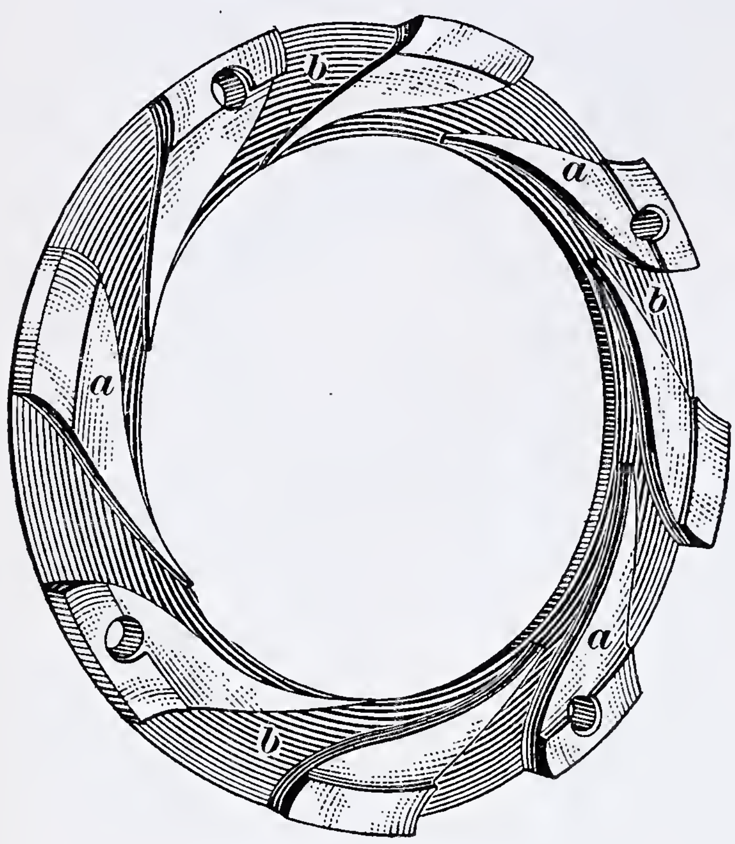


Fig. 28

37. Diffusion Rings.—A cross-section of one form of diffusion ring and its relation to the impeller is given in Fig. 17 (*b*), while a perspective view of a similar ring is given in Fig. 28. In the latter figure, *a* are the diffusing vanes that form the expanding passages *b*, which are marked *b* and *o*, respectively, in Fig. 17 (*b*). Diffusion rings are usually made of bronze, and in order that the efficiency of the pump may be maintained and their purpose fulfilled, great care is necessary in their design and construction.

ECONOMIC CONSIDERATIONS

38. Advantages of Centrifugal Pumps.—The centrifugal pump has won favor among pump users because of its efficiency, simple construction, the absence of complicated mechanism, valves, or pistons, the small space occupied, its smoothness of operation, freedom from water hammer, shock, and vibration of pipe and attachments, its accessibility, completeness of lubrication, low cost of maintenance, etc. The centrifugal pump has a decided advantage over the reciprocating pump in that it is impossible to subject the

pump and piping to a higher pressure than that corresponding to the speed at which the pump is designed to be run. No harm, therefore, can result by closing the valve in the discharge pipe while the pump is in operation, for the impeller will merely churn the water.

Centrifugal pumps have no valves or restricted passages to hinder the flow of a liquid through them, and are therefore serviceable not only for pumping hot or cold liquids but can be used also for pumping water containing large quantities of mud, sand, gravel, or anything else that can be carried through the pump and pipes by a current of water. Water containing much foreign matter in suspension cannot be handled successfully by reciprocating pumps or rotary pumps of the positive displacement type, as the solid matter would soon destroy the working parts or prevent closing of the valves. Centrifugal pumps of the volute type when used for sand dredging and similar work where the liquid to be pumped contains sharp quartz or any other material whose abrasive action would soon cut out a cast-iron pump, often have the casing fitted with a steel lining, which may be a manganese-steel casting, this material being so hard that it cannot be machined except by grinding. The impeller in such a case is made of the same material as the lining of the casing. In a lined centrifugal pump the lining may be replaced by a new one when worn out, this repair being cheaper than replacement of the whole pump.

For handling corrosive liquids centrifugal pumps of both the turbine and the volute types are often made entirely of special corrosion-resisting compositions, instead of being lined.

39. Head and Speed.—A centrifugal pump is usually said to produce a certain head rather than that it will pump against the head. Thus, a pump may be said to produce a head of 150 feet, which means that the pump will raise water 150 feet. The total head that can be pumped against, or the total head produced by a single-stage centrifugal pump, is determined by the velocity with which the water leaves the periphery of the impeller, and this velocity depends on the

diameter of the impeller and its speed of rotation. The peripheral speed does not usually exceed 160 feet per second or the number of revolution 4,000 per minute. Experiments have shown that the greater the speed the greater the efficiency of the pump for a given head and discharge, within the working limits of the pump; but if the speed is too great, the parts of the pump will have to be abnormally heavy to stand the strain, and the vibrations in the pump and its connections are excessive. This is particularly the case where small quantities of water have to be discharged against high heads, and if the head is high the quantity pumped should not be less than 75 gallons a minute. There is no doubt, in view of the impurities nearly always contained in mine water, that speeds above 3,000 revolutions per minute should be avoided, even for small pumps, although on pumps driven by steam turbines, speeds in excess of this are frequently used. Thus, boiler-feed pumps having a speed of 3,600 revolution per minute with a delivery of 500 gallons per minute against a pressure of 300 pounds per square inch are not uncommon. Such service is, however, of a specialized character, and perhaps permits of speeds that are not usually possible in other kinds of work.

40. Pump Characteristics.—A centrifugal pump works with the maximum efficiency when it delivers the quantity of water against the head and at the speed for which it was designed. If any one of these factors, quantity, head, or speed, is increased or decreased the efficiency of the pump becomes less.

As a rule, centrifugal pumps are stationary and during their lifetime pump, day after day, the same quantity of water against the same head at the same speed and hence run with the maximum efficiency. But there are numerous cases in engineering practice, as in shaft sinking, tunnel driving, foundation excavation, etc., where the quantity of water and the head vary within quite wide limits while the speed of the pump is constant. It is desirable, then, that when the work demanded of a pump is variable it be so designed that it has a reasonable efficiency over as wide a variation in discharge and

head as possible. The capabilities of any pump are best determined by actual tests, and the results may be quickly interpreted by plotting what is called a *characteristic curve*, often called a *head-capacity curve*. Such a curve for a constant-speed pump is shown in Fig. 29. To construct such a curve lay off, on squared paper, the discharge in gallons per minute horizontally and the heads against which the pump is working vertically on the left. In the figure, each horizontal space is equivalent to a discharge of 50 gallons per minute and each left-hand vertical space is equivalent to a head of 25 feet. As the tests are made, the heads are plotted against the corresponding capacities, after which the head-capacity curve

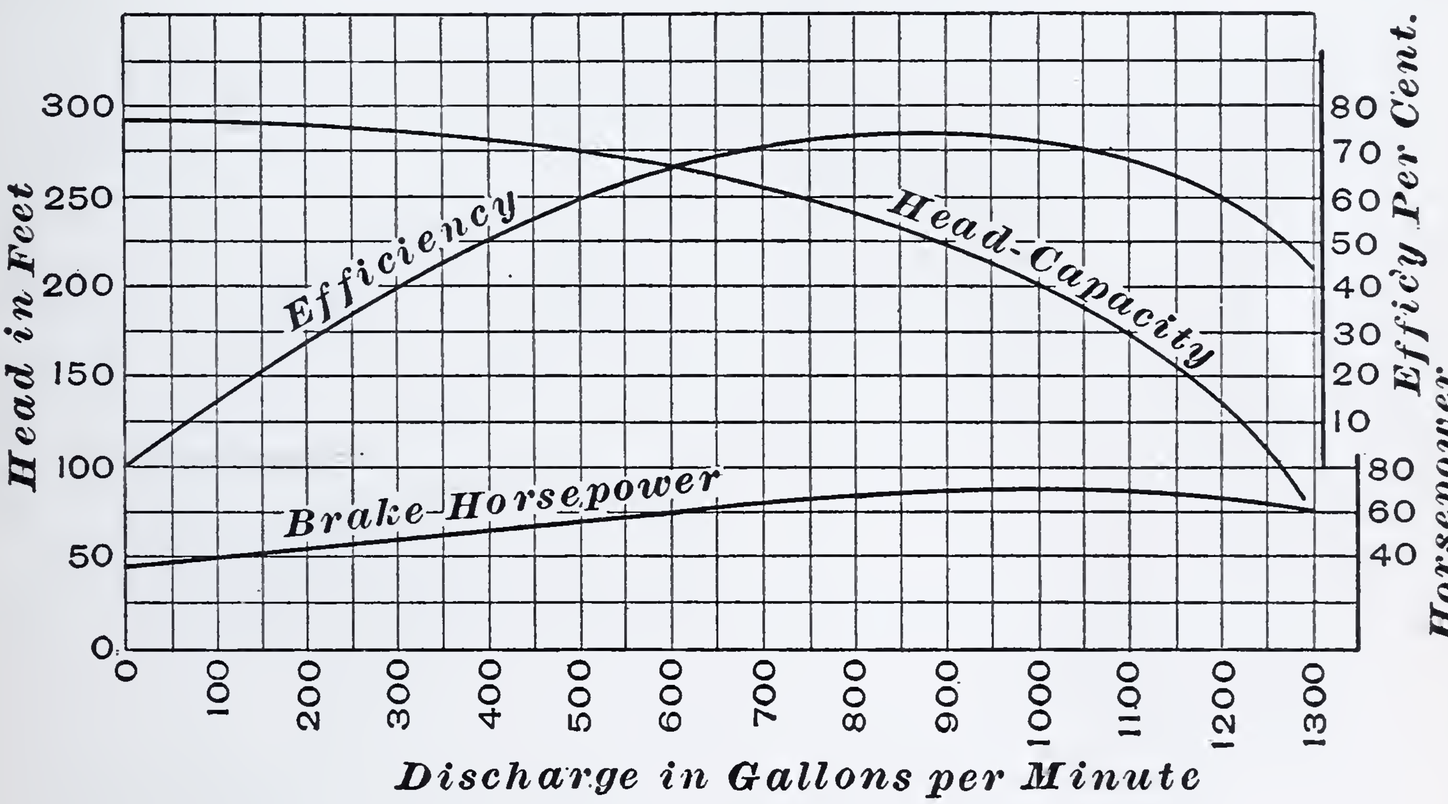


Fig. 29

is drawn in. At the same time, the efficiency and brake horsepower at each test can be plotted resulting in the efficiency and brake-horsepower curves, respectively, the values for which are marked vertically on the right. It will be noted, by an inspection of the head-capacity curve, that as the volume delivered increases the head decreases, and that after the volume reaches a certain amount, the decrease in head compared with volume occurs quite rapidly. Likewise, it will be noted from the efficiency curve that the efficiency between about 60 and 72 per cent. is maintained over a considerable range of head and capacity. Similarly, the brake-horsepower curve shows a nearly uniform power requirement

through the efficient field of operation of the pump. The limits of this efficient field of operation, as shown by the head-capacity and the efficiency curves, are 500 gallons at a 275-foot head with an efficiency of 60 per cent., and 1,000 gallons at a 200-foot head with an efficiency of 71 per cent. This pump cannot be economically used beyond these limits for when pumping, say, 200 gallons a minute against a 290 foot head the efficiency is only about 27 per cent.; and beyond a

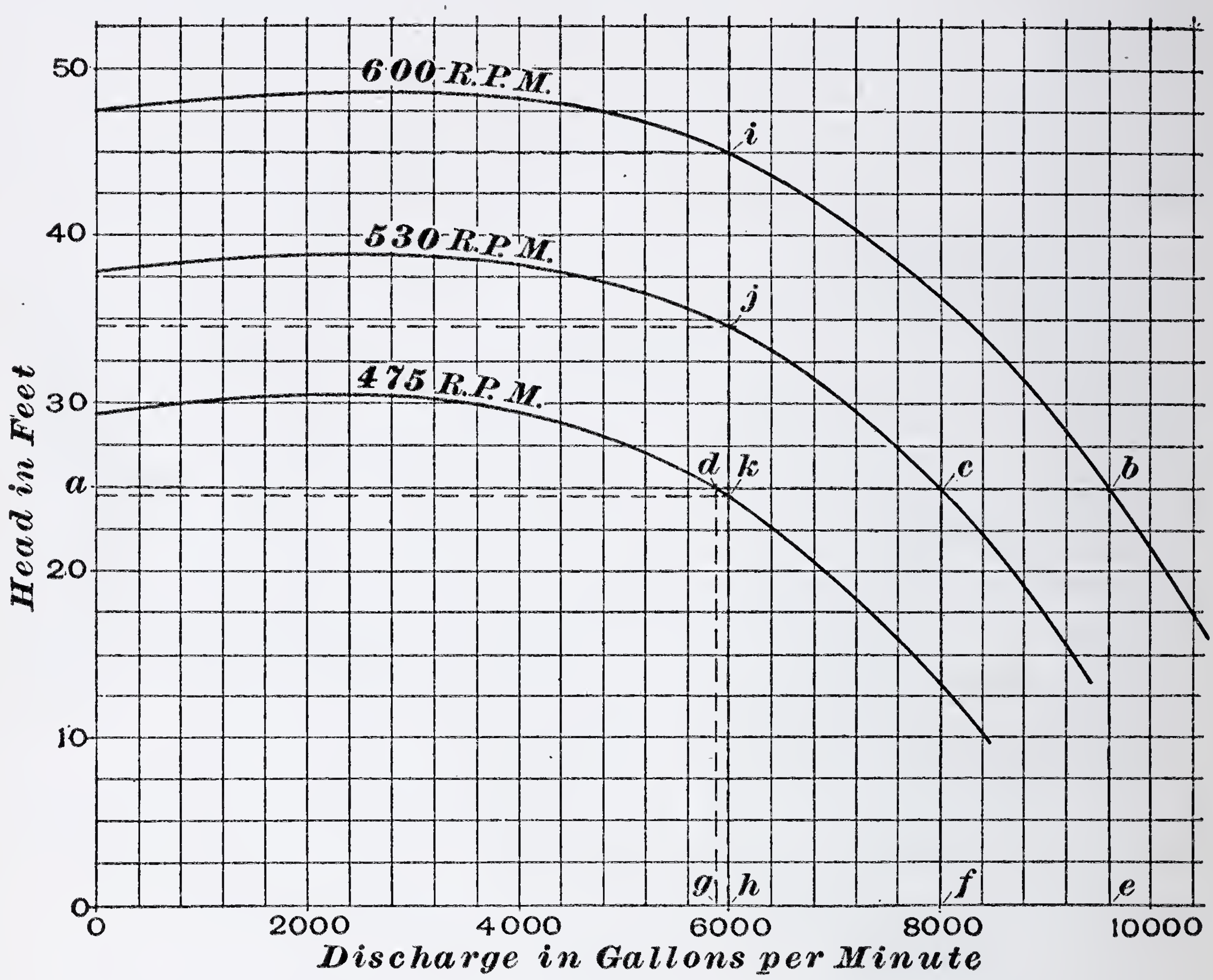


FIG. 30

capacity of 1,200 gallons per minute against a head of 140 feet the efficiency drops rapidly.

41. It is frequently stated that if the head against which a centrifugal pump is working is changed the capacity will necessarily change at the same time. This statement is correct only in the case of a pump driven by a constant-speed motor. If a variable-speed motor is used, a variation in capacity at constant head, a variation in head at constant capacity, or a variation in both head and capacity can be obtained. These possibilities are graphically illustrated in

Fig. 30, which shows the head-capacity curves for the same pump when operating at 475, 530, and 600 r. p. m., respectively. Horizontal and vertical lines projected from any point on one of the curves to the scales at the lower and left-hand margins, will give the capacity and head, respectively, at the speed marked on the curve. For illustration, assume that the pump whose head-capacity curves are charted in Fig. 30 is to give a varying capacity against a head of 25 feet. A line *a b* is drawn from the 25-foot mark at the left of the diagram until it intersects the three curves, and from the points of intersection, *b*, *c*, and *d*, vertical lines *b e*, *c f*, and *d g* are drawn to the lower margin. The capacities at the different speeds, and under a 25-foot head, are then indicated by the points *e*, *f*, and *g* on the lower scale, which correspond to 9,600, 8,000, and 5,850 gallons per minute, respectively. By a similar process, a constant capacity may be selected, say 6,000 gallons per minute, and by drawing a vertical line *h i* from 6,000 to intersect the three curves at *i*, *j*, and *k*, and then from these points drawing horizontal lines to the left-hand scale, the various heads available at this fixed capacity are found to be about 24.5, 34.5, and 45 feet, respectively. By plotting intermediate curves, other possibilities can, of course, be indicated.

42. Comparison of Volute and Turbine Pumps.—The simple volute pump is mechanically more efficient when handling large volumes of water at low heads, and the turbine pump is more efficient when handling small or moderate volumes of water at high heads; and this efficiency may reach 76 per cent. or a little more under favorable conditions of volume, head, and temperature. It has been shown by tests that for heads of less than 130 feet the volute pump is more efficient and that for greater heads the turbine pump gives better results; and this applies to multistage as well as to single-stage pumps. However, in practice the use of single-stage volute pumps is usually limited to 100 feet head, particularly where the water is dirty or gritty, as at mines. Single-stage turbine pumps, in capacities of about 1,000 gallons per minute, have been operated successfully at sufficiently high speeds to produce

a head of 250 feet. At a speed of 1,440 revolutions per minute, a head of about 170 feet per stage is considered good practice where the water pumped is clean, but where gritty or acid water is handled it is customary to limit the head per stage to 140 feet and to use speeds not exceeding 1,750 revolutions per minute with 60-cycle current, and 1,440 revolutions per minute with 25-cycle current. When the head exceeds 150 feet it is customary to employ stage pumps in any case.

43. Selection of a Centrifugal Pump.—A centrifugal pump should be of durable construction, its parts should be accessible, and its performance should be equal to the demands of the service in which it is to be used. The high rotative speed and the great variety of liquids handled make it necessary that every part of the pump, including the shell, impellers, shafts, and bearings, shall be of sufficient strength to withstand high speed and rough service without undue deterioration. The design should be such that the interior parts of the pump, as well as the more vital exterior parts, may be quickly reached and replaced without having to disturb the driving apparatus. Not only should the pump be capable of performing, with the highest possible mechanical efficiency, the particular duties for which it was designed, but it should also be able to do effective work, without affecting the pump, its driving mechanism, or mechanical efficiency, when the head, quantity of water delivered and, to a certain extent, the speed are varied within reasonable limits.

44. When determining the proper size and type of pump for a given service, it must be remembered that belt-driven pumps can be operated at any reasonable speed; hence, in their case, the problem involves merely the selection of a pump of the required capacity and of a sufficient number of stages to work easily against the prescribed head.

For direct-connected pumps, however, where the speed is fixed, full descriptions of the conditions to be met are necessary, as well as details covering the source of power and data regarding the type and kind of driving apparatus. Therefore, to obtain a pump suitable for operating at a fixed speed, the

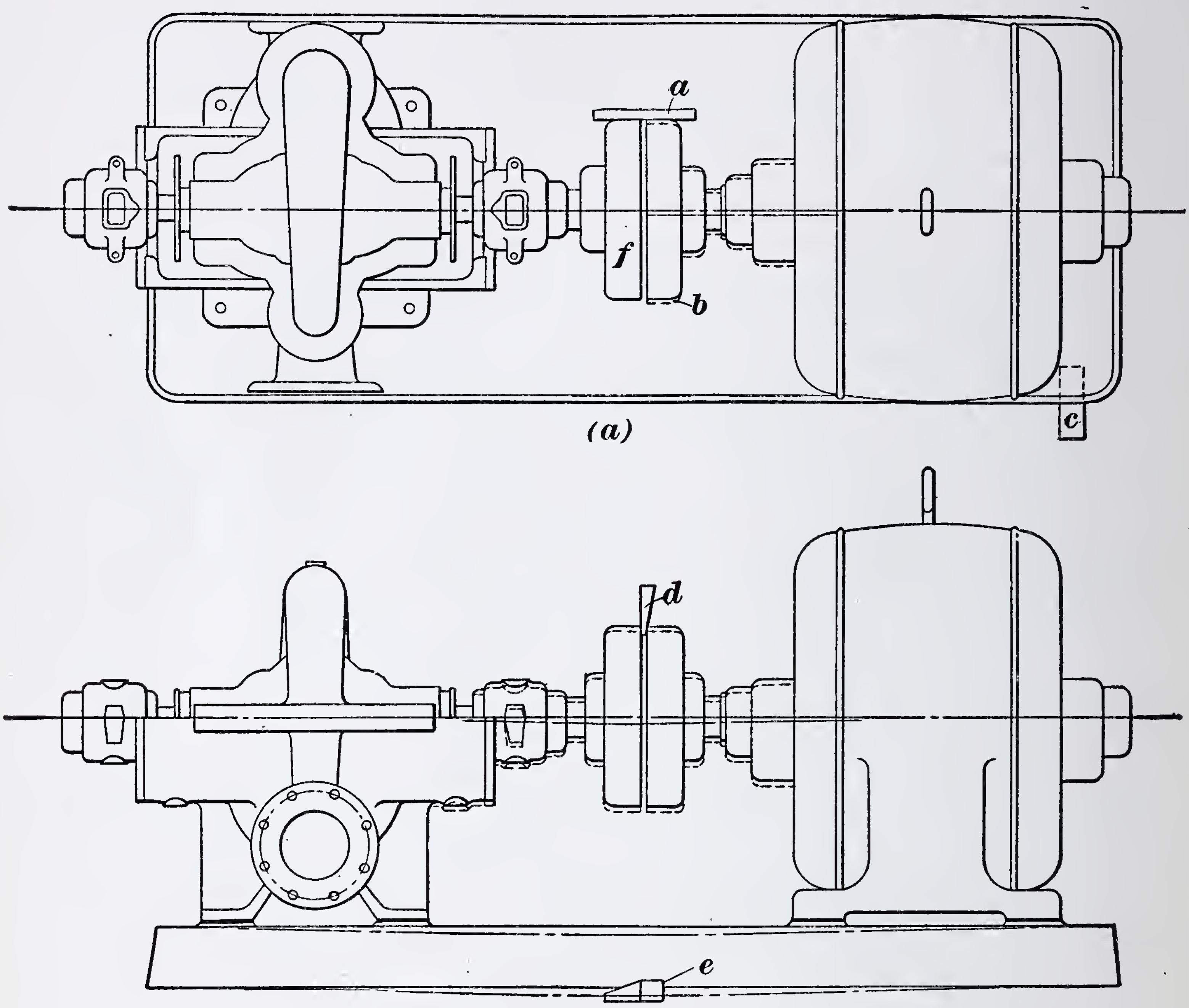
following information should be furnished the pump manufacturer: The quantity of water to be pumped; the kind of water, whether salt, acid, gritty, hot, cold, etc.; a sample or an analysis in case the water is acid; the suction lift and distance from the supply, together with diameter and condition of the suction pipe, and the number of valves, bends, and strainers; whether the suction and discharge heads include the friction head; whether the service is continuous or intermittent; the position of the suction and discharge openings; and the kind and amount of power available.

PUMP INSTALLATION AND OPERATION

45. Location of Pump.—The pump should be placed where it is reasonably accessible for attendance and repairs. The base plate should be somewhat above the floor line, and this may be assured by building a substantial brick or concrete foundation. The pump should be so situated, if possible, that water will flow to it from the sump and, in any case, the height of the pump inlet above the surface of the water in the sump should not exceed the maximum suction lift of the pump.

46. Lining Up Pump.—After the pump has been placed on its foundation, it is necessary to check the alinement of the pump shaft and the driving shaft. In the case of a direct-connected pump, the checking of the alinement may be accomplished through the use of a steel gauge applied to the connecting coupling, as shown at *a*, in Fig. 31 (*a*). When the parts are lined up correctly, the gauge *a* will fit evenly against the edges of both flanges all the way around. If the flange *b* is slightly out of line, as indicated by the dotted outline, the error can be greatly reduced by raising the lower corner of the base plate, using a wedge as shown at *c*. A test should then be made to see that the distance between the faces of the flanges is everywhere the same; which may be done by the use of the wedge *d*, as shown in Fig. 31 (*b*). If the bed plate does not bear fully on the foundation, it is well to reinforce it by inserting a wedge as at *e* to prevent sag which will throw

the motor and pump out of line and cause them to assume the positions shown by the dotted lines in (*b*). When correct alinement has been secured, the base plate may be grouted in; but not until the grouting is in place and the alinement has been checked again should the coupling bolts be put in the flanges *b* and *f*. Chisel marks should then be made on each half of the coupling directly opposite each other, in order to



(*a*)

(*b*)

Fig. 31

identify the position in which the coupling is to be replaced after making repairs.

47. Pipes and Valves.—The suction and discharge piping should have as few bends as possible. The suction line should be as short as possible, is usually one size larger than the discharge pipe, and must be absolutely air-tight, particularly if the pipe is very long or the lift great.

An effective check valve should be placed in the discharge line between the pump and the gate valve, and it is advisable

to install a foot-valve in the suction pipe to facilitate priming. When a foot-valve is used, the gate valve should be closed before the power is shut off. The end of the suction pipe should be protected by a strainer and, this, in turn, should be protected by enclosing it in a chamber which has removable screens, accessible from the surface of the sump. When pumps

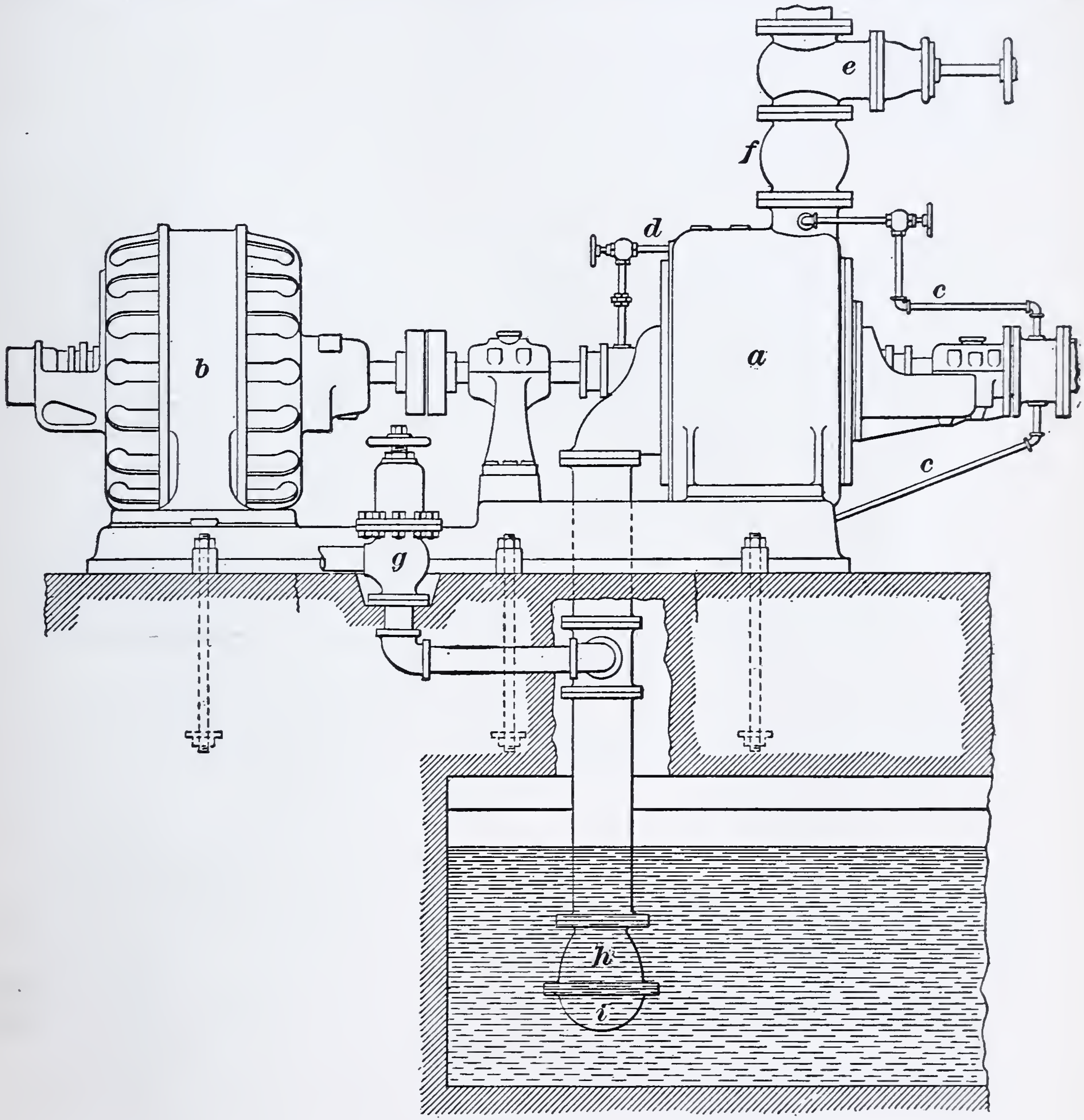


Fig. 32

are used against high heads, a water relief valve should be placed in the suction line adjacent to the pump. The pump should be drained to prevent it from freezing during periods of idleness in cold weather, and for this purpose the necessary drains should be provided.

48. The general arrangement of the piping, valves, etc., of a centrifugal pump is shown in Fig. 32. In the illustration,

a is the pump, *b* the motor, *c* the water cooling pipes, *d* the water seal, *e* the gate valve, *f* the check valve in the discharge pipe, *g* the automatic relief valve, *h* the foot valve, and *i* the strainer in the suction pipe.

49. Priming by Gravity.—All centrifugal pumps must be primed before they will deliver water. Where the pump is set below the level of the sump and the water flows directly into the pump with sufficient head to fill the casing, no other provision for priming is necessary; but where there is a suction lift, as is almost always the case, some provision must be made for priming the pump before it is started.

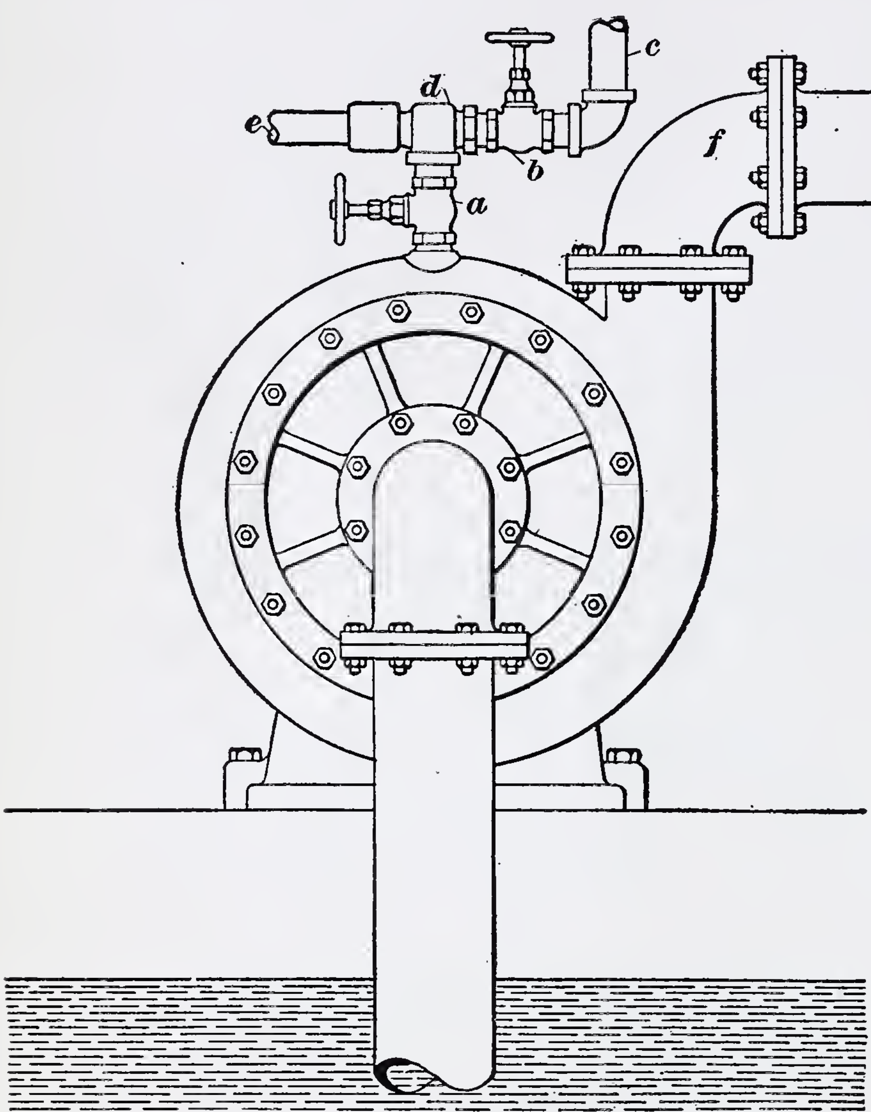


Fig. 33

50. Priming by Foot-Valve.—When the suction line is provided with a foot-valve it will act as a check and prevent the priming water from running back into the sump. Water for priming may be obtained from the discharge line, if that is full, or from any available source and, entering the top of the pump, will fill the casing. All pet-cocks or air vents on top of the pump should be opened during the priming process to allow the air to escape from the interior of the pump. Such means of escape for the air should be provided at the high point of each stage of a multistage pump.

51. Priming by Vacuum Pump.—If it is impracticable to obtain priming water from the discharge line, or if for any reason a foot-valve cannot be used, provision should be made

for exhausting the air from the pump casing either by a hand pump or by a power-operated wet vacuum pump.

52. Priming by Exhauster, or Ejector.—Where steam or compressed air is available, priming may be done by using an exhauster, or ejector, attached to the highest point of the pump casing, as shown in Fig. 33. By first opening the valve *a* and then the steam valve *b*, the flow of steam from the steam pipe *c* through the ejector *d* and out through the waste pipe *e* will draw the air from the casing and the suction line, provided that a tight-fitting valve is installed in the discharge pipe *f* close to the pump. When water flows from the waste pipe *e*, it is evident that the pump is fully primed, whereupon the valve *a* and then the valve *b* should be closed in the order named.

Another method of using an ejector for priming is shown in Fig. 34. Here the water does not enter through the suction pipe of the pump, but is lifted by the ejector *a* from the sump *b* through a separate pipe *c* by suction, and delivered into the top of the pump casing through the valve *d*. A foot-valve *e* on the end of the suction pipe of the pump is necessary to prevent the water from running back into the sump. When the pump is primed, water will appear at the air vent *f*, then the valve *d*, and the valve *g* on the steam supply pipe *h*, may be closed.

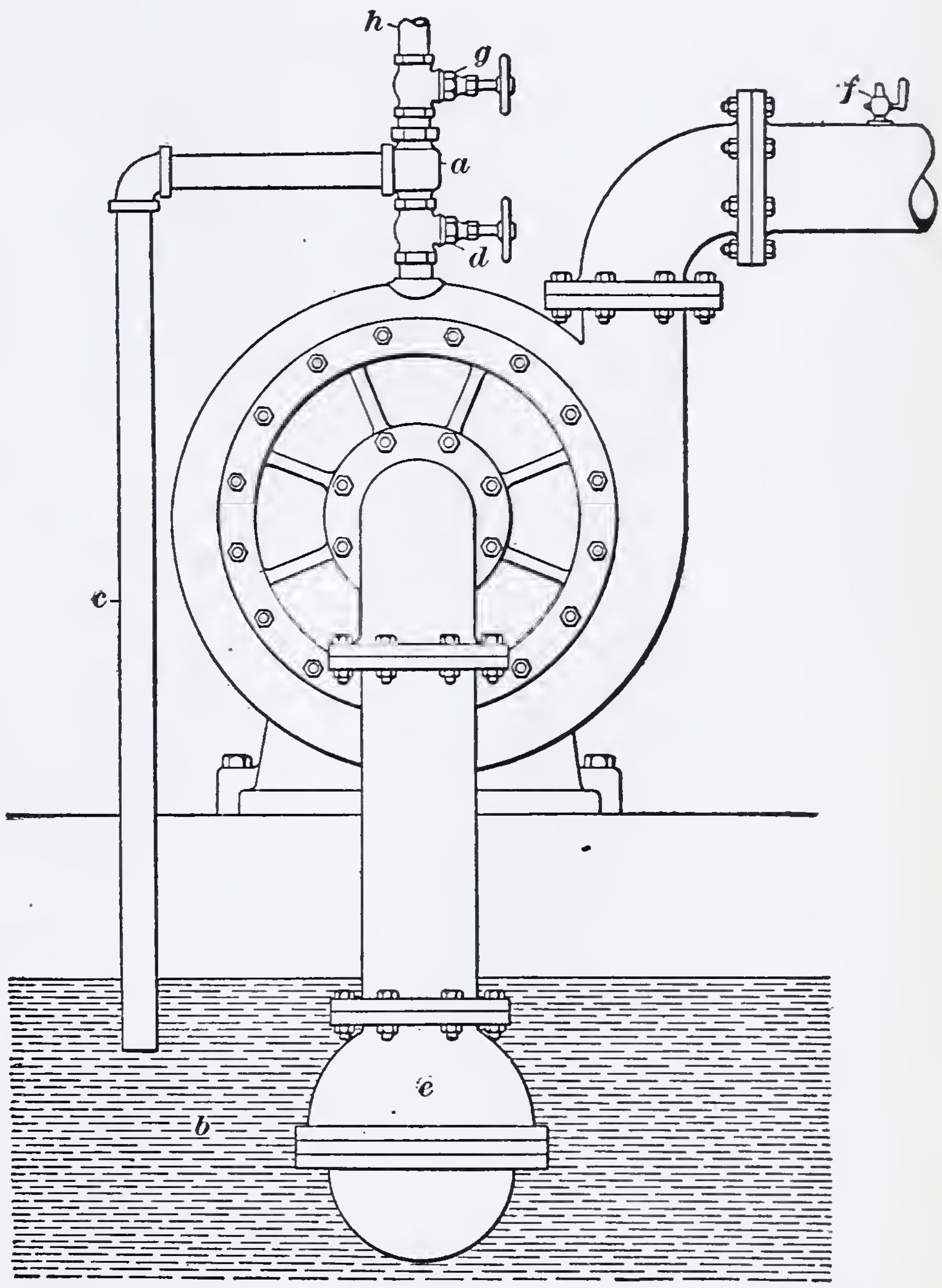


FIG. 34

53. Precautions in Starting.—Before starting a new pump the bearings should be carefully examined to be sure that they are free from pieces of metal or other substances that may not have been removed at the shop or may have gotten into the bearings while the pump was being installed. They should then be lubricated with a suitable grade of oil. The water-seal valves should be opened before the pump is started, and the water used for sealing should be clean; as mine water is frequently acid and gritty, this may necessitate a separate source of supply. The stuffingboxes must be cleaned and packed, and enough packing placed back of the water-seal ring so that the sealing water enters through the ring and not through the packing. The pipe supplying the sealing water should be air-tight; otherwise air entering at the sealing ring might cause the pump to loose its suction. A centrifugal pump must not be started or run without being primed, because the clearance rings or labyrinth rings have a very small clearance and are liable to bind, heat, and cut and, thus, be ruined if the pump is run without water. The direction of rotation of the motor should be the same as that of the pump as stamped on the casing, for pumps must not be run backwards. One of the characteristics of the centrifugal pump is that when primed and operated at full speed with the discharge valve closed, the horsepower required to drive the pump is considerably less than that required to drive it when the pump is working against its rated head with the discharge valve open. Therefore, it is much to the advantage of the driving apparatus, particularly where alternating-current induction-motor drives of large size are used, to keep the discharge valve closed when starting and open it when the pump is running at full speed. This procedure is also advantageous for shunt-wound direct-current motor drives as it relieves the motor of a heavy starting duty.

When the pump is in operation, air may be freed from the water and collect in the passages of the pump. For this reason, the air valves on top of the casing should be opened from time to time; or these valves may be left continuously part way open, and the small amount of water discharged

from them drained away through the proper connections. Should the discharge diminish when the pump is running under normal conditions, the pump should be opened and the valves and impellers examined for the presence of extraneous matter, such as chips, pieces of coal, etc., which may be the cause of the trouble.

In shutting down the pump, the gate valve in the discharge pipe should be closed before the motor is stopped. The pump may then be drained, if necessary, by opening the drain cocks at the bottom of the casing and the air valves at the top.

54. Pump Sizes.—The size of a centrifugal pump is expressed by giving the diameter of its discharge opening. Thus, a No. 5 pump is a pump in which the diameter of the discharge opening is 5 inches. This method of designating pump sizes is purely arbitrary and gives no idea of the actual dimensions of the pump or its capacity and, as the manufacturers do not agree among themselves, a No. 5 pump made by A may have a very different capacity from a No. 5 pump made by B.

Formulas have been suggested for calculating the size, or number, of a centrifugal pump to discharge a given volume of water against a given head, but they have not proved applicable in practice. The only way the discharge of a centrifugal pump can be determined is by actual measurements. When the discharge of a pump of a given size has been determined by measurements, a formula may be devised that will give the discharge of other pumps, provided the ratio of the parts in the other pumps is the same as that in the pump tested. It is apparent that such formulas are not of general application, but are confined to the pumps of the same manufacturer; that is to say, each manufacturer will have a formula that is applicable to his pumps, but not to those of another pump maker.